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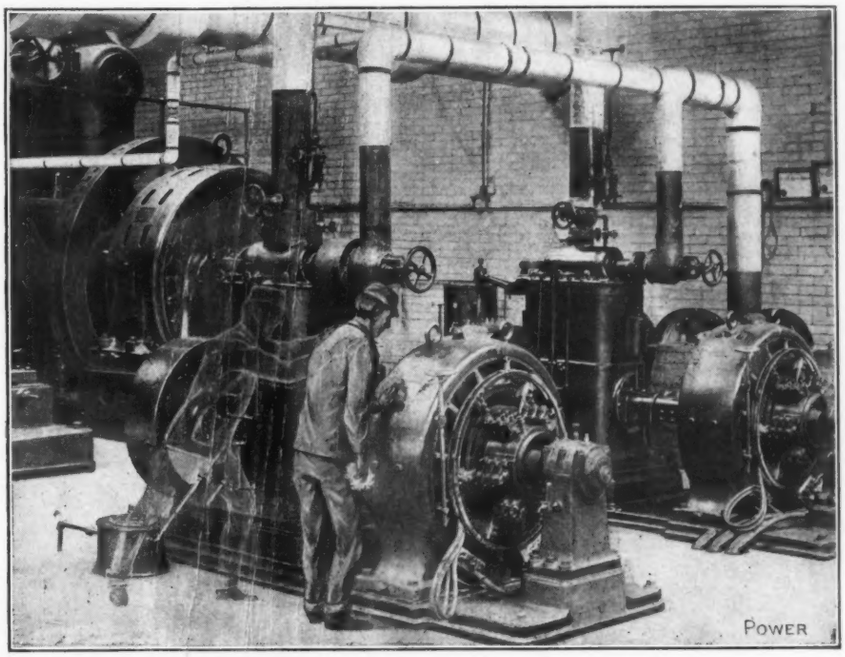
Forcing Oneself to Work

UNFORTUNATELY many men engaged in power plant operation are unfitted either physically, mentally or by temperament for the work. Others are not naturally qualified or do not possess the necessary training to make good operators.

Such men labor under either a physical, mental or nervous strain. Their work is not a pleasure to them. They lack enthusiasm and must force themselves to perform their duties. They soon fall into a rut and content themselves with a commonplace existence in the lower ranks.



Every man possesses certain qualifications which make him better suited for one line of work than for another. A young man just out of high school finds himself at the door of the power plant, seeking work. If he makes a fair appearance and if help is needed, he is given a chance, doing some unimportant task. If he is faithful and shows natural ability, he is advanced as his experience, training and conditions warrant. If his ability and trustworthiness are merely commonplace, he becomes a mere operator. If he shows no ability or liking for the occupation, he is discharged after a fair trial.

The same young man might have become the driver of a laundry wagon, a molder's helper or a house painter. His selection of occupation was haphazard, as was also his selection for the position.

That a man may get the most out of his work it must be congenial to him and he must be suited to it. To raise the standing of steam engineers it is necessary to raise the standard of the personnel. Every down-in-the-mouth, illfitting and illfitted stationary engineer lowers the general efficiency of the whole a corresponding amount. Any organization of workmen recognizing this fact and endeavoring to improve the standard of the individual workmen is bound to ultimately raise the standard of workmanship.

Many men are content to work in a half-hearted, indifferent manner, earning a mere living at work which is not attractive to them because their enthusiasm has never been aroused and they have never come to realize the value and importance of their work.

Important and well-paid positions are to be filled in the power plant field, and men having a vision of something beyond the doors of the engine or fire rooms will be chosen to fill the positions worth having.

If a man is physically unsuited to his work, he had better try something else. If he merely lacks enthusiasm he should get in touch with the various sources of inspiration, such as the national and local organizations and the leading technical journals dealing with the work. If he lacks training and experience, he had better get busy and be ready for the worth-while position when it comes.

Central Power Plant of Montgomery, Ala.

By Warren O. Rogers

SYNOPSIS—The main features of this installation are the manner of coal delivery, piping of the generating and auxiliary units and the design of the pump house, the latter made necessary by the great difference between the low- and high-water levels of the river.

In Montgomery, Ala., two electric-generating stations compete for the electrical business of the city. As a result, the rates charged for current are low.

What was known as the Citizens Electric Light Co. is now the Montgomery Light & Traction Co. Strange as it may seem, the rival electric company supplies the energy for the street-car service, this being one of the provisions of a long-term contract between the Citizens Electric Light Co. and the rival company. It seems, however, that this part of the contract has not been carried out and so poor was the service, that the new plant, described herein, was built by the traction company to not only supply electrical energy for commercial use in the city, but to also operate its own street-car lines. The matter is now being decided by the courts as to the validity of the contract, the traction company maintaining that it has been broken.

Power Plant

This new power plant is on the outskirts of the city on the bank of the Tallapoosa River, and has several interesting features. The boiler house is 175 ft. long by 75 ft. wide, and the turbine room is 100 ft. long by 110 ft. wide. The machinery is arranged in symmetrical order and so spaced that there is room to work on each unit without interference with the next. The building is in two sections, so to speak, the boiler room running at right angles to the turbine room. Between the turbine room and the

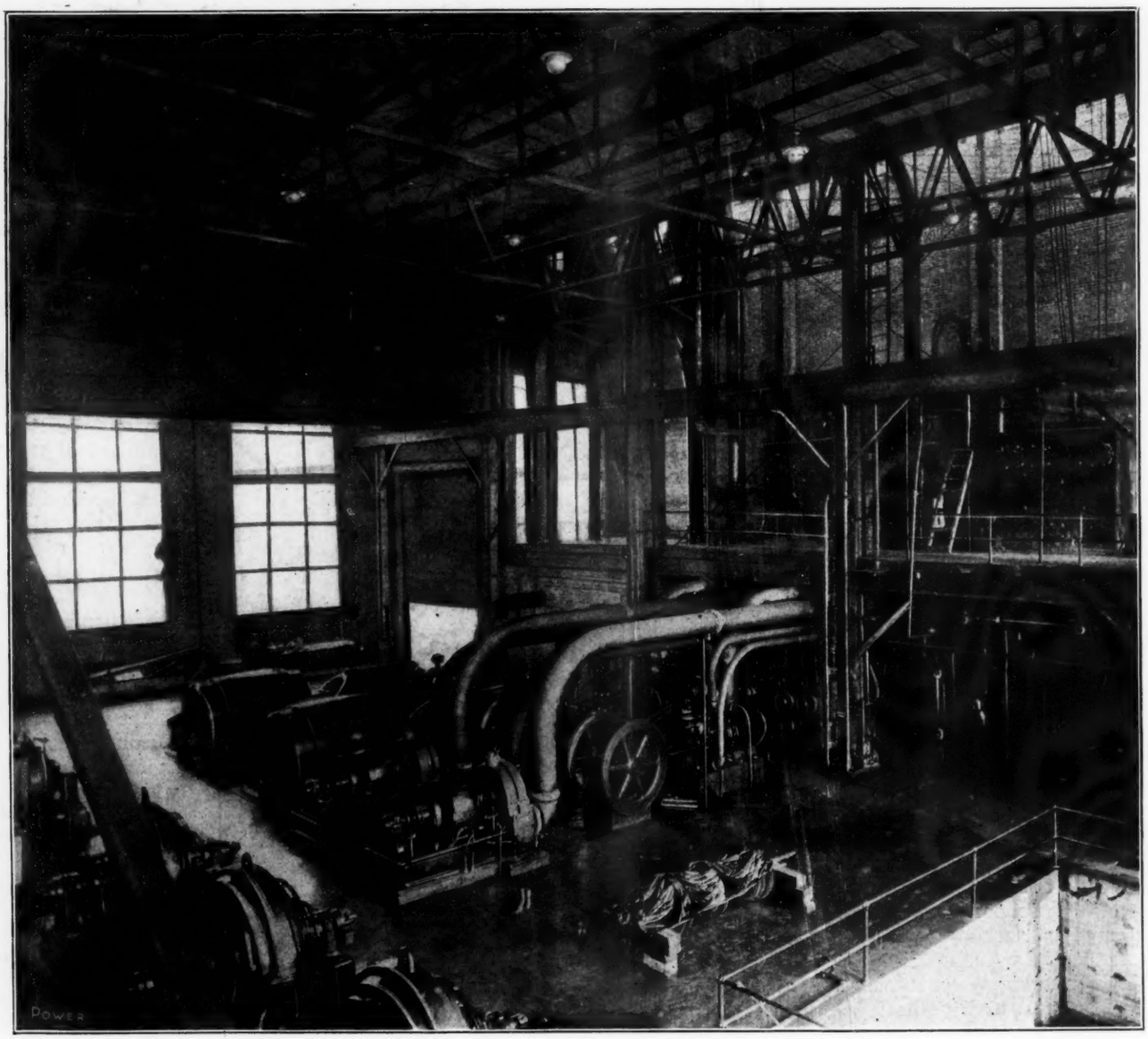


Fig. 1. View of the Turbine Room from the Switchboard Gallery

brick boiler-room partition is a gallery and pump bay. On the other side of the turbine room, is a second gallery and bay; both galleries running across the room. The floor of the turbine room slopes toward the center from four sides. A wide-gage track enters the turbine room at one side and crosses to the engineer's office, which is on the switchboard side of the room on the ground floor. A 20-ton crane runs the length of the turbine room and heavy material can be unloaded from the freight cars on the track and deposited at any desired point. A small 10-ton crane runs the length of the gallery next the boilers.

Generating Units

Two 2500-kv.-a. horizontal steam turbines drive the two 2500-kv.-a., 2300-volt, two-phase, 60-cycle generators at 3600 r.p.m. They are placed side by side with the steam end next to the boilers, as shown in Fig. 1. On the side of the room next to the generators, and between them and the switch-bay, are two 200-kw., 1200-r.p.m., rotary converters, changing the alternating current to direct current at 500 volts for power circuits through the city. There are also three 500-kw. rotary converters, running at 900 r.p.m., delivering 600-volt direct current.

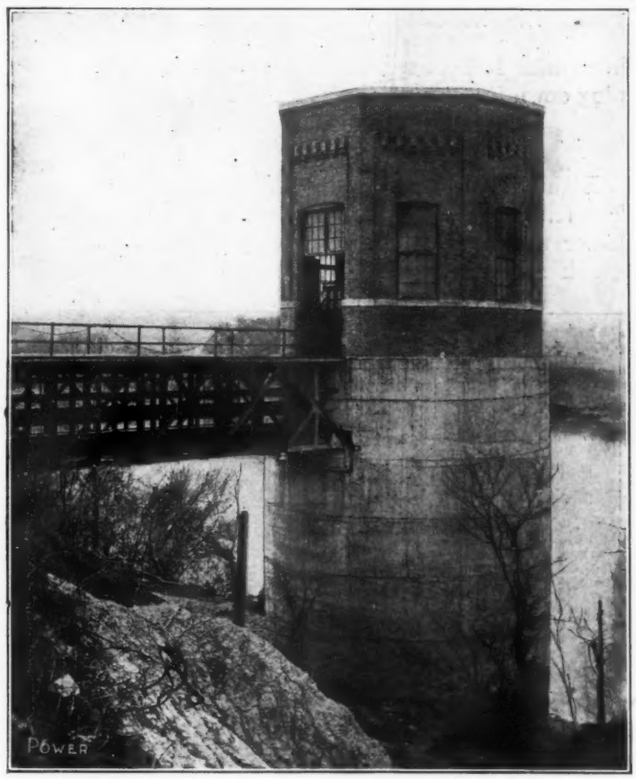


Fig. 2. Pump House and Concrete Pit

Two turbine-driven exciters are at the end of the dry-vacuum pumps. The turbines are directly coupled to 100-kw., 125-volt, direct-current generators, of 800 amp. capacity, at 2400 r.p.m.

Condensing Outfit

On the turbine-room floor are two 10 and 24 by 18-in. steam-driven air pumps, which are piped to two barometric condensers placed next to the boiler-room wall. The exhaust connections from the turbines run under the floor to the condenser, as do the circulating-pump discharge pipes from the two condensers. As water in abundance is obtained from the Tallapoosa River, with an average temperature of 60 deg. F., the rise in temperature is but 5 deg. with a 29-in. vacuum. In fact, the water from the condenser is used to cool the oil used on the turbine bearings.

Pump House

Owing to the height to which the water in the river rises above the normal level, it was necessary to construct

Fig. 3. Interior of Pump House, Showing Motors

a pump house and pit on the river bank. The pit is 65 ft. deep and 25 ft. in diameter. The lower part is round and built of reinforced concrete with 3-ft. walls. Above the concrete work the walls are brick and hexagonal in shape. The pump house is illustrated in Fig. 2.

The circulating pumps are at the bottom of the pit and are of 5000 gal. capacity per min. against a 100-ft. head, at low water, and 8000 gal. capacity per min. at high water. The variation in the level of the river between low and flood stages is about 60 ft. These pumps are directly connected to two 150-hp. induction motors, using 2200-volt, two-phase, 60-cycle current, and running at 1165 r.p.m. They are shown in Fig. 3.

An iron ladder extends from the floor of the pump house to the concrete bottom. Gate stands connect with the discharge-valve stems, which facilitate the operation of the valves. Beside the motor-control apparatus, there is a gage board, upon which are mounted eight gages for indicating the suction- and discharge-water pressures. At the back of the pump pit is a small counterweighted elevator used for lowering light loads to the bottom of the pit. A small motor-operated, four-ton, pivoted, single-arm crane extends from the wall of the pump house to the center, which can be moved to any position desired. Speaking tubes run from the top to the bottom of the pit, also to the switchboard gallery in the station. A railing surrounds the opening in the flooring over the pump pit.

Pumps

Except the two centrifugal pumps in the pump house, all of those in the plant are in the pump bay, which is under the condenser and heater gallery. They include

two 17 and 10 by 15-in. duplex, outside-packed pumps, used for boiler feed, and two 8 and 10 by 12-in. duplex pumps, used for house service and for pumping the hot-well water to the two 5000-hp. open heaters. The heaters are on the balcony above the pumps. Exhaust steam from the auxiliaries enters the heaters at the end, or can be bypassed to the atmospheric exhaust. The heaters can be operated independently of one another. The feed water is heated to from 190 to 200 deg. The locations of the pumps, heaters and condensers are shown in Figs. 1 and 4.

The latter illustration also shows the arrangement of the steam and exhaust piping, which is suspended from the gallery flooring. Steam from the boiler is conveyed to No. 1 turbine with a long and short bend at right angles to each other. No. 2 turbine is supplied from the header by a steam line extending from a tee to the turbine with a long-radius bend. The steam pipes to the dry-vacuum pipes are connected in a similar manner. The exhaust-steam lines connect with a 16-in. exhaust-steam header, which runs the length of the pump bay and connects either with the heater or with the atmospheric exhaust pipe.

Steam pipes, bent to the shape of a question mark, drop from the steam header to the pumps. This construction permits of free expansion and contraction. The peculiarity of the auxiliary piping is shown in Figs. 1 and 4. Both live- and exhaust-steam pipes are run in a horizontal line at right angles from the headers, dropping down to the several units as shown.

EQUIPMENT OF THE MONTGOMERY LIGHT & TRACTION CO.'S POWER PLANT.

No.	Equipment	Make	Purpose	Kw.	Phase	Cycles	Volts	Amp.	Hp.	Pres. lb.	Size	R.p.m.	Manufacturers
2	Turbines	Horizontal	Main units	2500	...	..	..	...	..	200	..	3600	Westinghouse Machine Co.
2	Generators	Alt. current	Main units	2500	2	60	2300	543	..	...	..	3600	Westinghouse Electric & Mfg. Co.
2	Converters	Rotary	Power circuits	200	2	60	500	333	..	...	..	1200	
3	Converters	Rotary	Railway	500	2	60	600	833	..	...	..	900	
2	Generators	Direct current	Exciters	100	...	..	125	800	..	...	..	2400	
2	Turbines	Horizontal	Exciters	100	...	..	..	...	..	200	..	2400	Westinghouse Machine Co.
2	Condensers	Barometric	Main units	...	...	..	..	...	..	...	..	..	Henry R. Worthington.
2	Pumps	Dry air	Condensers	...	...	..	..	...	..	...	..	..	
2	Pumps	Circulating	Cooling water	...	...	60	2200	...	150	...	5000 gal.	1165	De Laval Steam Turbine Co.
2	Motors	Induction	Cir. pumps	...	2	60	2200	32	150	...	..	1165	Allis-Chalmers Co.
2	Pumps	Duplex	Boiler feed	...	...	..	..	...	..	...	17x10x15"	..	Henry R. Worthington.
2	Pumps	Duplex	House service	...	...	..	..	...	..	...	8x10x12"	..	
2	Heaters	Open	Feed water	...	...	..	..	...	5000	...	..	..	Harrison Safety Boiler Works.
8	Boilers	Stirling	Steam gen.	...	...	..	..	...	400	200	..	..	Babcock & Wilcox Co.
4	Transformers	Oil cooled	Station rotaries	100	1	60	2310–370	...	..	...	..	..	Westinghouse Electric Mfg. Co.
6	Transformers	Oil cooled	Lighting	250	1	60	2310–370	...	..	...	..	..	

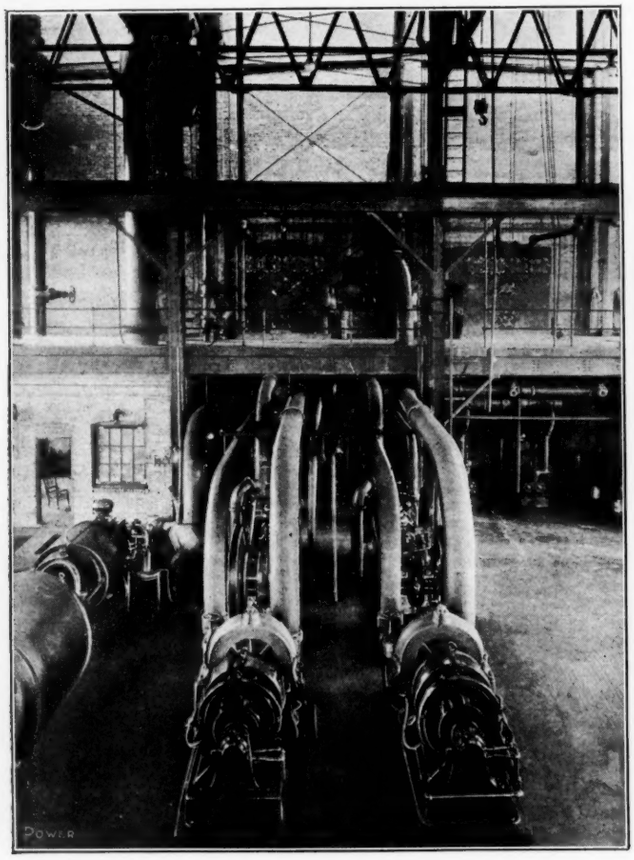


FIG. 4. SHOWING PIPING, PUMPS, HEATERS AND BAROMETRIC CONDENSERS

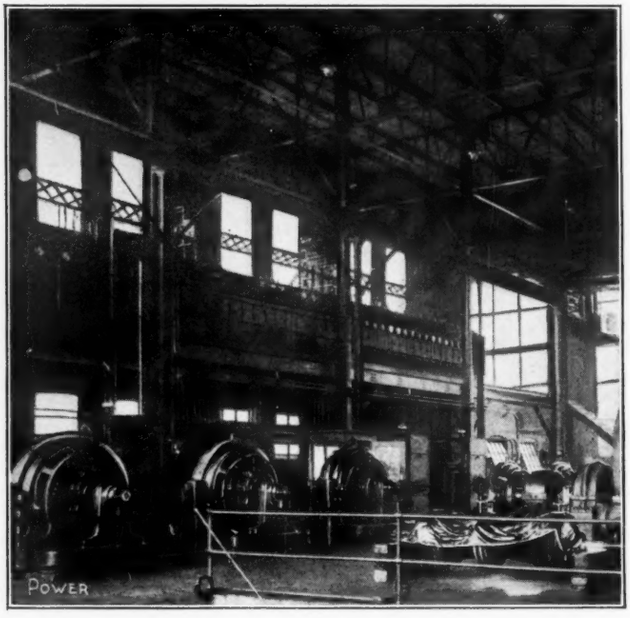


FIG. 5. ROTARY TRANSFORMERS AND SWITCHBOARD, FROM TURBINE FLOOR

BOILER HOUSE

Eight 400-hp., water-tube boilers, carrying steam at 200 lb. pressure are set in a row along one side of the boiler room. They are all equipped with superheaters, which increase the temperature 125 deg. Each boiler contains 290 tubes, 3¼ in. in diameter, three 25-in. steam drums and one 42-in. mud drum. The main 16-in. steam main runs the length of the building, 18 ft. above the floor level and in front of the boilers above the firing

to the 16-in. header through a 12 to 16-in. steam main in the boiler room. The steam-outlet pipes for the boiler connect with the header from the top with short-radius bends. The main header, above the pumps, is connected

alley. It contains two U-expansion bends to relieve the pipe line of expansion and contraction strains. Feed water is supplied through a 6-in. main suspended under the steam main, branch pipes running to the several boilers.

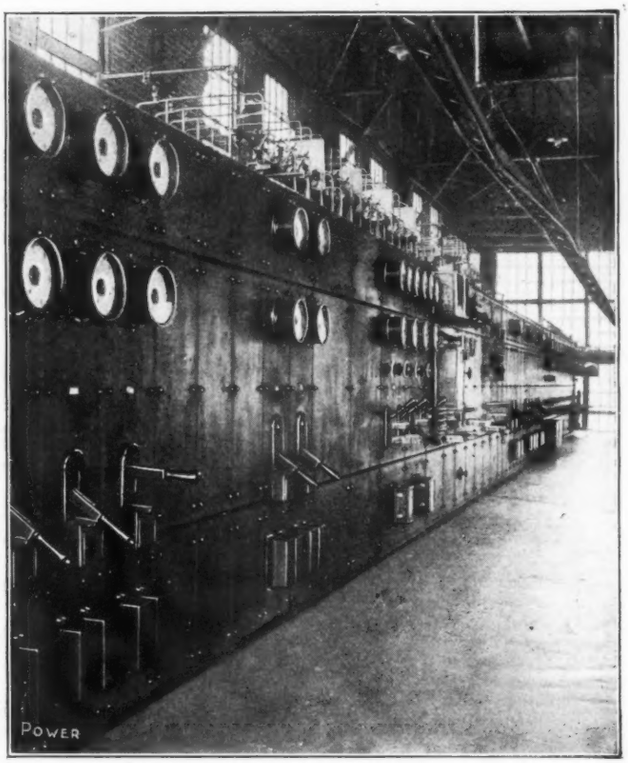


FIG. 6. SWITCHBOARD BALCONY

The grate area of each boiler is 6x14 ft., or 84 sq.ft. The furnaces are fitted with balanced doors, which are a protection to the fireman in case a tube bursts.

Coal is delivered to the boiler house in dongola coal cars and unloaded into a coal bin built with concrete partition walls and wooden sides. It has eight compartments and a total storage capacity of 480 tons. The concrete partition walls support a wide-gage track upon which the loaded coal cars are run to be unloaded. The boarded sides of the bin are high enough from the floor to permit coal to run down within easy reach of the fireman.

There is room for eight other boilers of the same size as those now installed, on the opposite side of the coal-storage bin. All smoke uptakes from the boilers enter a main smoke flue, which connects to a steel chimney 10 ft. inside diameter at the top, 25 ft. diameter at the bottom and 200 ft. high. At present, hand firing is used and the ashes are removed to the ashpit in front of the boilers, wheeled out and dumped at the rear of the station. Provision has been made, however, for an ash-handling system; an ash tunnel has been constructed and the ashpit made with a temporary bottom.

TRANSFORMERS

Along the side of the transformer-turbine room, next to the switchboard, are four single-phase transformers for station use. They are of 100 kv.-a. capacity, transforming from 2300 to 370 volts, 60-cycle current. There are also six single-phase transformers of 250 kv.-a. capacity, transforming from 2300 to 370 volts, 60-cycle, Fig. 5.

Above the transformer, suspended from the under side of the switchboard balcony, are the busbars. Below on the main floor are the oil switches and above on the balcony is the 58-panel slate switchboard, shown in Fig. 6. These panels are divided into groups, for the generating units, exciter units, motor and lighting units about the plant, instruments and commercial lights and motor-line control.

The plant is in charge of Chief Engineer J. M. Chumley, who has been with the company for more than 20 years.

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Determination of Steam Consumption

BY EARL B. SMITH

SYNOPSIS—A formula by which the steam consumption of a turbine or engine may be determined when exhausting into a jet condenser, having given the load, the condenser discharge and the steam and water temperatures.

❖

When a turbine or an engine is exhausting into a surface condenser, it is a simple matter to determine the amount of steam used. But when the exhaust is into a jet condenser, a great many difficulties are presented. The steam consumption in this case must be determined either by a careful boiler test or by computations from temperature and volume determinations at the jet condenser. In large lighting and railway plants the boiler-test method usually is not feasible because of the difficulty of isolating a part of the boiler plant for use on the test; then the drips, the leakage, and the auxiliaries usually offer other complications.

A determination of the steam consumption of a turbine or an engine from the condenser end, when the condenser is of the jet type, need not be very difficult. The following is a description of a method that may be used on turbine tests, which has been found by the writer to be very satisfactory.

In most cases it is more convenient to determine the amount of water discharged from the condenser. Of course, if the amount of injection water may be determined and also the amount of condenser discharge, then the difference in a given time gives the steam consumption. But usually the amount of condenser discharge is the only quantity that may be determined. Temperature readings of the injection water and of the condenser discharge must be taken at frequent intervals.

Let

W = Pounds per minute discharged from the condenser, which is the mixture of injection water and the condensed steam;

w = Pounds of steam per minute supplied to the turbine (wet);

$(W - w)$ = Pounds of injection water supplied to condenser;

H_s = Total heat per pound of steam, w, supplied; taking into account the quality of the steam ($= q + xr$);

Hw = Total heat per pound of condenser discharge;

H_e = Total heat per pound of steam exhausted from the turbine before entering the condenser;

h = Total heat per pound of injection water.

Then the fundamental formula obtained from the total heat relations is

$$(W - w)\ h + w\ H_e = W\ H_w$$

Therefore,

$$w = \frac{W\ (H_w - h)}{H_e - h} \qquad (1)$$

To make a practical formula of this equation other factors must be introduced which will account for the various losses of heat and work. Take the case of a 6000-kw. low-pressure Parsons turbine direct-connected to a 5000-kw. alternating-current generator running at 1000 r.p.m.

Let,

B = Brake horsepower of the turbine;

K = Load in kilowatts on the generator during test;

F = Bearing friction and windage of the turbine, which is approximately 1 per cent. of full load;

E = Efficiency of the generator; which is 97 per cent. at full load, 96 per cent. at ¾ load and 94 per cent. at ½ load;

L = Windage loss and brush friction of the generator in kilowatts, approximately 2½ per cent. of full load;

R = Heat units lost by radiation from the shell, etc.; approximately 1000 B.t.u. per hour per square foot of the shell (3000 B.t.u. per minute in this case);

S = Total steam consumption per hour = 60 w.

The difference between the heat received by the turbine and that exhausted represents the work done plus all the losses. Then

$$w\ (H_s - H_e) = 42.41^*\ (B + F) + R$$

and since

$$B = \frac{K + L}{0.746\ E}$$

$$H_e = H_s - \frac{42.41}{w}\left(\frac{K + L}{0.746\ E} + F\right) - \frac{R}{w} \qquad (2)$$

Substituting equation (2) in equation (1),

$$w = \frac{W\ (H_w - h)}{H_s - h - \dfrac{42.41}{w}\left(\dfrac{K + L}{0.746\ E} + F\right) - \dfrac{R}{w}}$$

Simplifying,

$$w = \frac{W\ (H_w - h) + 42.41\left(\dfrac{K + L}{0.746\ E} + F\right) + R}{H_s - h} \qquad (3)$$

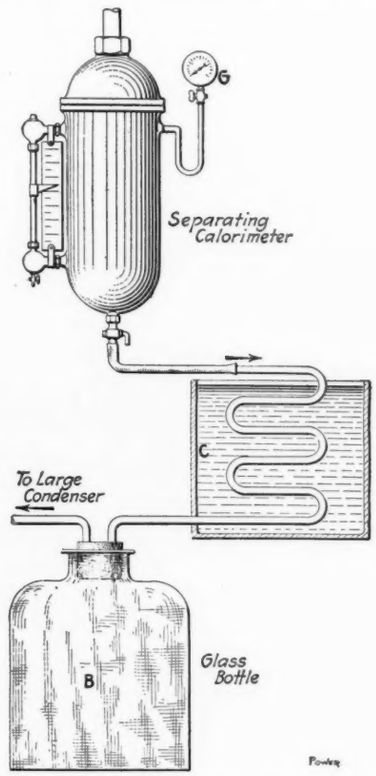
Then, since $L = 0.025\ K$ and $F = 0.01\ K$, substituting these and the numerical values indicated above, the following formula is obtained for the total steam consumption per hour at full load:

$$S = 60 \times \frac{\left(\begin{matrix}W\ (H_w - h) + 58.6\ K + (58.6 \times 0.025\ K) \\ + (42.41 \times 0.01\ K) + 3000\end{matrix}\right)}{H_s - h}$$

$$= 60\ \frac{W\ (H_w - h) + 60.49\ K + 3000}{H_s - h}$$

$$= \frac{60\ W\ (H_w - h) + 3629.4\ K + 180{,}000}{H_s - h}$$

Formula (3) is applicable to the case of any direct-connected turbine or engine unit exhausting into a jet condenser. The method of determining W, the amount of condenser discharge, depends upon the installation. In some cases a carefully calibrated weir may be used, or a pitot tube may be more convenient, or perhaps a venturi meter may be used.



METHOD OF COLLECTING CONDENSATE FROM CALORIMETER

The value of H_s, the total heat in the steam received by the turbine, must be determined by a temperature or a pressure reading in the steam pipe, and a quality determination of the steam. This quality of the steam must be determined by means of a steam calorimeter. If the steam used is under the usual high pressures, then a throttling calorimeter may be the more convenient. But if the steam is low pressure (below atmosphere), then a separating calorimeter should be used. In the use of the separating calorimeter with low-pressure steam the small gage, G, ordinarily used for indicating the amount of steam flowing through cannot be used; but a small condenser made of a coil of copper pipe, C, immersed in a tank of cold water may be connected to the exhaust of the calorimeter. The condensate from this small condenser must be collected in a closed vessel

*The value 42.41 is the heat equivalent of a horsepower-minute.

B, which is connected to the vacuum side of the turbine condenser. Then, knowing the pressure of the steam (or its temperature) and its quality, x, the total heat may be computed by the aid of steam tables and the formula $H_s = q + x\,r$.

Formula (3) may be simplified somewhat further by dropping the items L, F and R, and thus obtaining the formula

$$w = \frac{W\,(H_w + h) + 42.41\left(\frac{K}{0.746\,E}\right)}{H_s - h} \qquad (4)$$

This formula (4) gives results sufficiently accurate for all practical purposes and which will be within 0.4 per cent. of those of formula (3). Also it may be noticed that when using formula (3) great accuracy is not necessary when selecting values for L, F and R. Even an approximate value of E may be used.

Talbot Water-Tube Boiler

By T. H. Heath

A water-tube boiler with features that will appeal to the engineer and power producer is becoming well known on the Pacific coast. Besides safety and economy, its strong points are reparability and freedom from scale. It is manufactured by the Talbot Boiler Co., Seattle, Wash. Its general construction is shown from Figs. 1, 2 and 3. Fig. 4 shows one of the smaller types of the boiler supplying steam to an engine-driven electrical unit.

The steam-generating parts consist of a number of steel headers at the back of the boiler from which water-tube members extend horizontally toward the front, each consisting of an outer and inner tube. The outer tubes are screwed into the header at the rear end, and are welded closed at the front or free end. The inner tubes, concentric with the outer ones, are also screwed into the header, but connect with a separate chamber; the other end is open and extends to within about six inches of the closed end of the outer tube. The tubes, as regards the headers, resemble a porcupine boiler, only the inner tubes of the Talbot boiler provide the circulating medium for the water, Fig. 5.

All headers, identical for each size of boiler, are set horizontally, one above the other, across the back, except the two lower ones, which are set vertically, or nearly so, directly below the horizontal ones. The tubes from the horizontal header extend over the furnace, but those from the vertical headers extend each side of the furnace. Fig. 1 gives a good idea of the arrangement of the headers and tubes, although the tubes on the firebox side are not shown, being covered by the insulating material. The front, or free ends, may be seen at the lower left-hand corner of Fig. 3.

The method employed for supporting the parts is shown in Fig. 1, although in later types the headers are carried by saddles and the water-pipe inlets by supporting legs. The free ends of the tubes are loosely supported by the front casing which allows unrestricted expansion. Iron-plate casing is used, with magnesia-asbestos insulation.

Referring to Fig. 1, the inlet is at the right end of the top header at A. By the inside arrangement of the header the water is compelled to go through all its tubes, and passing out through B, loops around under the casing and enters the next lower header at C. From this header and tubes the water passes out at D, crosses over to the right of the next lower one, through its tubes and thus goes through all of the horizontal headers.

From the last horizontal header the course is through the pipe E to the lower end of the left vertical header and, rising through it and its tubes, crosses over and enters at the bottom of the right vertical header at F, thence to the steam pipe G.

In Fig. 5 is shown a section of a header and the way the water is directed through the tubes. The water enters the chamber H through the ground joint and flows through the inner tubes which discharge near the outer

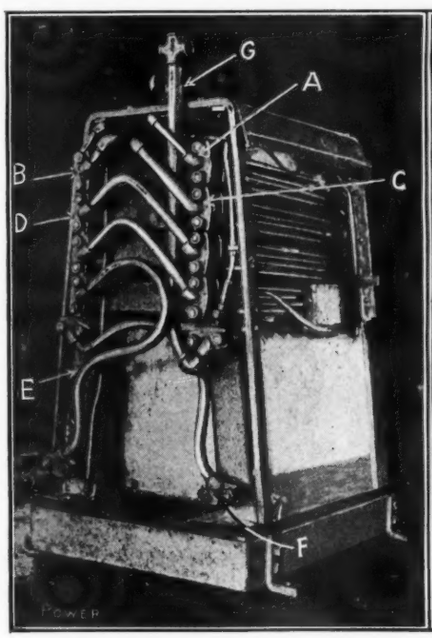


Fig. 1. Header Connections

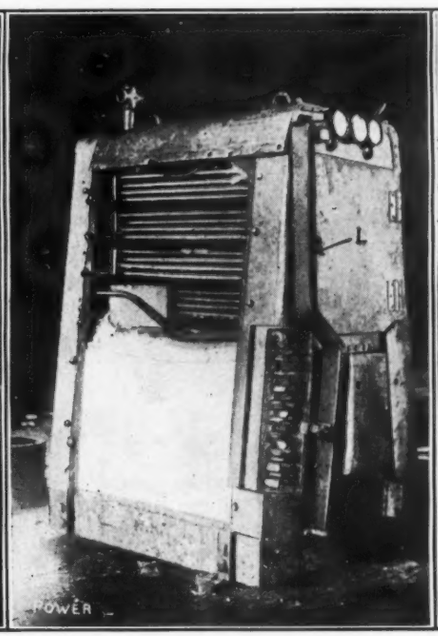


Fig. 2. Closed Tube Ends

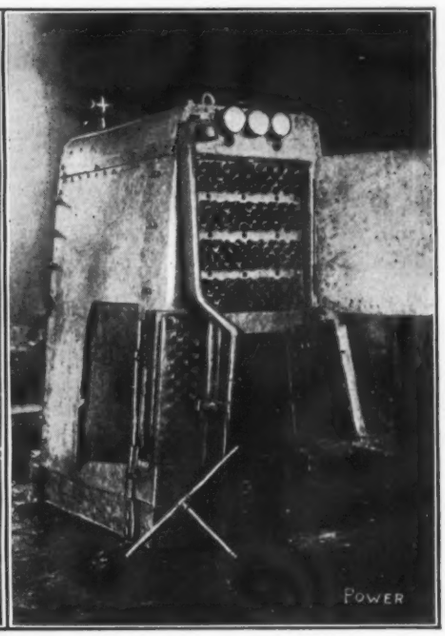


Fig. 3. End View of Boiler

end of the outer tube. The water returns between the two tubes to the chamber *K*, which passes it on to the next chamber connected with inner tubes only. Thus the water is directed back and forth over the heated zone, through four sets of tubes and headers.

It will be observed from Fig. 1 that the water-pipe connectors between headers are bent or looped upward. This is for the purpose of trapping or retaining the water in the headers when the boiler is not in operation.

The feed water is controlled automatically. The free end of one of the tubes is connected to a lever, and by its lineal contraction and expansion reduces or increases the flow of water by causing the lever to close or open a

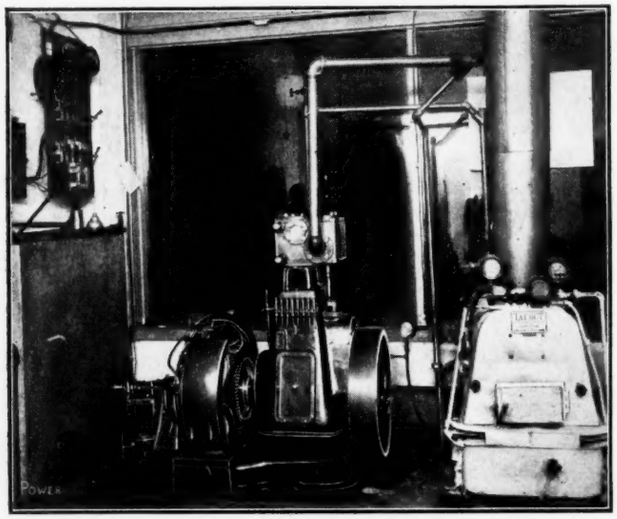


FIG. 4. BOILER AND ENGINE UNIT

piston-admission valve. An increase of temperature beyond a predetermined point expands the tube, causing the lever to open the admission valve, letting in more water; a lowering of temperature creates the reverse action to decrease the water supply. *L*, Fig. 2, shows the lever with the tube attachment. Water is delivered to the valve under constant pressure, generally from a pump, and bypasses the surplus back to the tank or supply; the same lever can be used to control the fuel oil when oil is used.

As the stack is directly over the top header tubes the feed water enters the coolest section of the boiler and works down into the hotter zone. Saturated or superheated steam may be used as desired. The degree of superheat is controlled by the water admission; reduction of flow causes an increase of superheat. The ordinary water gage or glass is not used in this type of boiler, the amount of water being indicated by a temperature gage—in appearance something like a steam gage.

There is only small steam-storage capacity in this boiler, as it is designed to generate steam as it is used. As a full head of steam can be raised from a cold boiler in from four to six minutes there need be practically no standby losses.

The headers are of crucible steel with cored tube chambers or passage ways and are tested to 1000 lb.; the tubes are of standard lap or butt-welded iron pipe screwed into the headers. On the small boilers, which have smaller pipes than the larger types, the inner tube is turned off on the outside, decreasing the diameter to give the proper clearance between them and the outer tubes. As they are subjected to a balanced pressure this does not interfere with the pressure capacity of the boiler. The ground joints between the headers and water connections, secured with studs, are shown in Figs. 1 and 5.

The rate of flow varies from a velocity of 3000 ft. in the smaller sizes to about 10,000 ft. in the larger types of boilers. The rapid circulation as well as increasing the heat-absorbing capacity also prevents pitting and formation of scale; boilers operated under working conditions after months of use are found unaffected.

Scale-forming solutions have been tried under normal conditions with the same results. The water is carried along so rapidly that there is no opportunity for the scale-forming elements to precipitate or adhere. In boilers freshly opened after service, the edge of the tubes screwed into the headers is found slightly rounded and polished by the rapidly moving sediment.

One of the important features of the boiler is its reparability. About the only thing that might possibly fail in the course of ordinary operation would be an outer tube; this could be considered an ordinary pipe-repair job. The free ends of the outer tubes are welded to standard shape to fit a spanner for screwing them in, and the inner tubes have a slot across the ends to fit a screwdriver arrangement; these operations may be done by opening the boiler front, when the tubes are at once accessible.

The casing is in renewable sections bolted to the supporting frame. Long fiber asbestos is pressed lightly over the tubes and the space between it and the casing is filled with magnesia.

As to safety, about the only thing that could happen would be a burst tube, and this would be no worse than

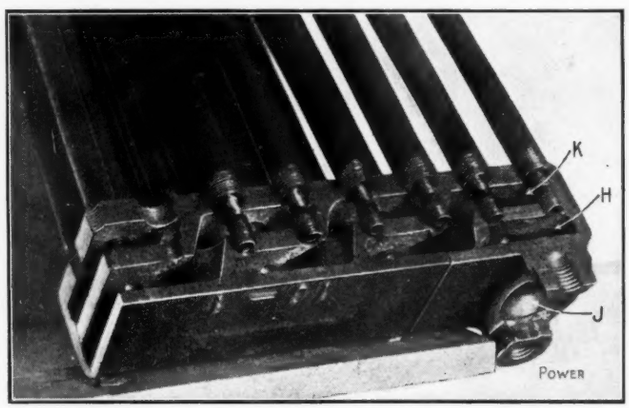


FIG. 5. DETAILS OF TUBE CONNECTIONS TO HEADERS

the bursting of a steam pipe, and there is no large volume of scalding water back of it. A header might let go, but this would be improbable, owing to its construction and large factor of safety.

Following is the result of a two-hour evaporation test that was made with a Talbot boiler rated at 100 hp.:

Water evaporated, lb.	7446
Oil used, lb.	490
Average temperature of feed water, deg. F.	97.4
Average temperature of feed steam, deg. F.	390
Average temperature of feed oil, deg. F.	65
Maximum stack temperature, deg. F.	350
Minimum stack temperature, deg. F.	265
Temperature of boiler room, deg. F.	70

The calorific value of the crude oil used was 18,500 B.t.u. per lb. and when corrected for moisture the test shows an evaporation of 17.31 lb. of water per lb. of oil.

The Rational Utilization of Coal

By F. E. Junge

SYNOPSIS—Inferior grades of coal which would hardly be worth transporting to industrial centers, are used at the mouth of the mines in coke ovens and producers. The gas is used in gas engines for producing high-tension electric current and innumerable byproducts are obtained from the tar residues. This practice is common in Germany.

As a result of geological retrospection and economic argument we are forced to the conclusion that all resources which are capable of utilization must be utilized, and that all values which are hidden in the soil must be recovered, in order to attain a desirable state of national efficiency. Especially coal, the greatest material asset of a country, should no longer be wasted by burning it on the grate, but, if it contains byproducts capable of recovery, it should be converted into its constituents, so as to yield the maximum possible returns in heat. Likewise, it is of great economic importance to utilize the large tracts of lignite and peat land, particularly those located near large undeveloped bodies of ore, and to render useful the inferior grades of coal, so abundant in mining centers. This may be done either by transforming the raw coals into some form of available energy which can be transmitted over long distances at reasonable cost, or by refining the low-grade coals into superior products, such as briquettes, coke, or chemicals, that they may serve as a basis for other industries.

The question which remains then is not whether the inferior grades should be used but how they can be used most efficiently. Before answering this question it is opportune first to get a clear idea of the meaning of the term "low-grade coal." There is no standard of designation which can be referred to and none that can be established. A fuel cannot be classified strictly by its relative heat value, nor by the volatiles it contains. The constant change in byproduct values or in the returns realized therefrom make it impossible to define clearly the limits below which a coal becomes inferior.

If, owing to their low carbon, high moisture and high ash contents, lignites and peat are referred to as low-grade fuels, traditional custom is being followed rather than plain facts based on recent developments. Likewise, there are conditions under which the screenings or smaller sizes of a high-grade lean coal may rank equal or lower in monetary value. It is only refuse such as culm and other waste, obtained in large quantities in coal mining and which hitherto escaped utilization entirely, owing to their excessive ash contents, that can be rightly classed as low-grade coals, since both their contents of fixed carbon and of volatile hydrocarbons is small.

For efficient utilization of coal there are now available the byproduct coke oven, the gas retort and the gas producer.

These have stood the test of practical experience and their exclusive adoption for the conversion of coal into superior products is no longer a matter of debate but of time and local consideration. The method of utilization depends on the heat value and chemical characteristics of the coal. The higher its heat value and the greater the transportation facilities available the larger will be the radius of commercial distribution of the coal from the pit. If the intrinsic value of the particular grade of coal is not high enough to pay its transportation charges it must be used on the spot, for the chief cost of coal at most places is not that of mining but of transporting it to the market. Hence, if by some mode of transmission either in the form of gas or electric current the transportation factor can be eliminated, the coal business is relieved of one of the chief factors which decrease profits, and prohibit commercial independence.

If the coal is of poor quality, such as culm, slack, refuse, screening and other waste containing a high percentage of ash, making it unmarketable, it may be convenient to discharge the screenings, without any attempt at their separation from dirt, slate, etc., upon a conveyor leading to the boiler room and use it for the generation of steam power. In some cases in Germany it is found that 30 per cent. of the mine output is worthless for selling, but if used at the mouth of the pit for generating electric energy, it is possible to sell power at a cost no greater than in a hydro-electric plant of the same size.

Sometimes, owing to excessive ash contents in culm and waste banks, these materials cannot be burned under boilers because the ash acts both as a diluent and as an obstruction to the combustion process. Nor can the materials be dumped back into the mines because of the danger of spontaneous ignition. Hitherto they were either stored in large piles in the neighborhood of the pit, or where territorial limitations prevented this, they were transported by rail into neighboring dumping grounds, being thus absolutely useless and causing heavy expenditure. Now there are two possibilities for using these low-grade coals. One is to dump them into Jahn's ring producers where their heat value is utilized. The 25 or 30 per cent. of combustibles in the culm yield a gas free from tar and well suited for heating, lighting and power purposes. A plant of this type has been in active operation for ten years in the von der Heydt coal mines, Saarbrücken, Germany. The cost of 1000 B.t.u. in the form of producer gas is only 0.005c. Or one brake horsepower-hour in gas engines costs 0.05c.

Another method was developed by Dr. N. Caro, of Berlin, Germany. It is based on the observation that culm banks and other waste contain more nitrogen than that which corresponds to their coal contents. In Westfalian collieries it was found that culm banks, the coal contents of which show on analysis of about 1.2 per cent. of nitrogen, contain up to 1 per cent. of nitrogen, although their total contents of combustible matter is only 25 or 30 per cent. Dr. Caro has succeeded in gasifying this material in producers of the Mond type, especially equipped for the purpose, and besides getting a suitable gas he gains about 80 per cent. of its total nitrogen contents in the form of sulphate of ammonia. At the same time, the sulphur is removed so that the residues of the gasification process can be directly dumped from the producer back into the mines without fear of causing ignition. From 60 to 90 lb. of sulphate of ammonia are gained per ton

of culm so that not only the cost of removing the coal is recovered, but a good profit is also realized.

There are places where coal of very small size is available in large quantities and at low cost, such as coal dust in coke and gas works. Their high ash and dust contents and the small size makes them unfit for boiler fuel, because the fine dust clogs up the flues and necessitates frequent cleaning. Nor are they suited for transportation. Two ways for utilizing these coals are now open. One is to burn them in especially designed gas producers, the other is in briquettes.

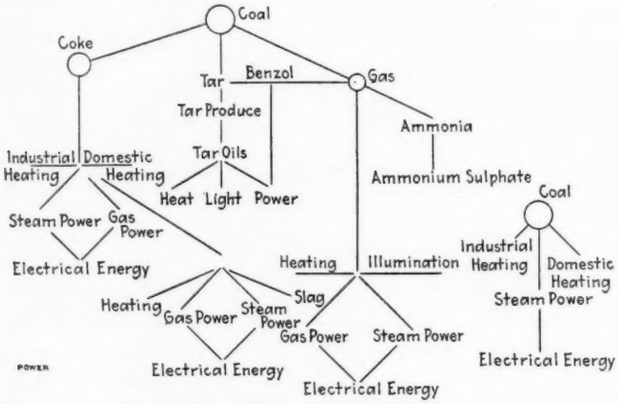


FIG. 1. COMPARISON BETWEEN COMMERCIAL POSSIBILITIES OF BURNING COAL IN A BOILER AND ITS GASIFICATION IN COKE OVENS

There are some points to consider when using dust coals in gas producers. The great resistance offered to the passage of the air must be overcome by keeping the charge as low as possible and uniform in height; otherwise the air will pass up along the walls, producing clinkers and a bad quality of gas. The coal must be charged frequently within short intervals and in small quantities, and if it contains moisture it must be preheated by the gas. This exchange of heat will increase the calorific value of the gas, at the same time lowering its temperature.

Under certain local conditions the smaller sizes, such as buckwheat, pea, dust anthracite, and dust coke, have a reasonable chance of competition and may even be superior to bituminous coal, lignite and peat fuels. For example, in a certain city gas power was to be adopted, and there were three different classes of fuel available, anthracite nut at $5 per ton, anthracite dust at $1.70 per ton, and lignite briquettes at $2 per ton, having the respective heat values of 13,500, 12,600 and 8100 B.t.u. per lb. The fuel consumption, including all losses, worked out at 0.25, 0.089 and 0.18c. per b.hp.-hr. for the three classes. In this particular case dust anthracite was superior, and it had the special advantage over lignite that it yielded a gas absolutely free from tar. Taking into consideration the factor of transportation it was found that the cost of fuel for operation with these three classes is equal when the plant is located at the following respective distances from the mine: anthracite nut 0, anthracite and coke dust 404, and lignite briquettes 73 miles. It is seen that dust coke and dust anthracite, owing to their low price and comparatively high heating value, have the largest commercial distribution radius, in this case.

There are other cases where dust fuels cannot be disposed of at a profit and where it is better to convert them into a more convenient form of fuel by briquetting. In one case in Riga large piles of dust coke were available, which had been sold previously as filling material for ceilings, bringing 2.5c. per 100 lb., while coke in the larger sizes sold at 30c. Although the dust coke contained from 75 to 80 per cent. combustible it was unfit for use in boilers, because of the objections previously mentioned. Therefore, a briquetting machine was installed which produced 1000 briquettes per hour. An addition of 5 per cent. of hard pitch and tar residues as binding material gave sufficient cohesion. The average production in a 10-hr. day was 8240 lb. of briquettes having a heat value only 5 per cent. lower than the coke, the higher ash content being compensated by the greater heat value of the tar and pitch. By the adoption of superior methods of utilization the returns from this low-grade material have been increased from 55c. realized per ton of coke dust to $3 received per ton of coke briquettes.

Summarizing, it may be said that coals of low heat value and others containing a high percentage of ash should be converted into electric energy at the mouth of the mine, the current being transmitted to neighboring districts, provided that there are markets within reaching distance. With the high voltages now employed there is, in small countries like Germany, practically no limit in distance, as far as the transmission of electric energy is concerned.

Coals of high heat value but of inconvenient form should

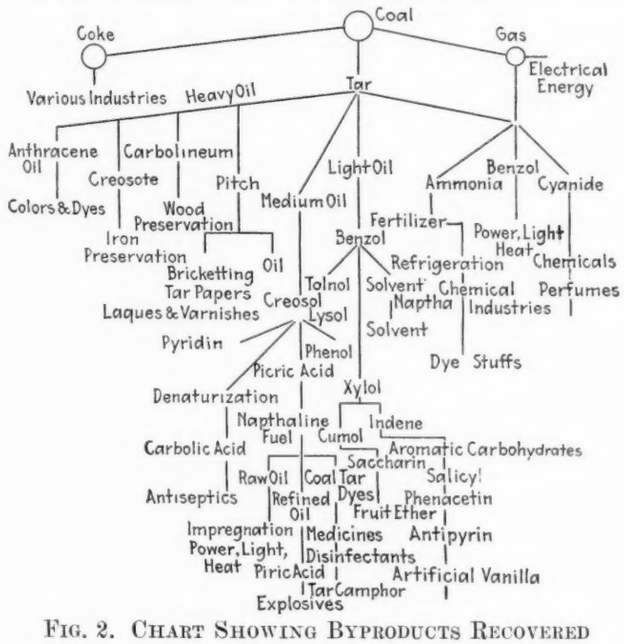


FIG. 2. CHART SHOWING BYPRODUCTS RECOVERED FROM COAL TAR

be either gasified in special producers and the power distributed by electricity, or they may be formed into briquettes, thereby increasing their value and rendering them capable of competition with the high-grade coals. Only those nonbituminous coals, such as anthracite and semi-anthracite, which contain a high percentage of fixed carbon and a low percentage of volatile hydrocarbons, and are convenient to handle, should be shipped to markets and used for direct combustion on the grate. Other conditions being equal, heat value and transportation fix a definite economic limit beyond which a certain coal from

a certain mine is no longer marketable and must be used at the mine.

The diagram, Fig. 1, gives a comparative analysis between the industrial possibilities inherent in the direct combustion of coal on the grate and in the gasification of coal in coke ovens. In the first case the application is limited to the generation of heat, steam power and electric energy. The applications in the second case are numerous and support a variety of industries; also the resulting coke can be used for heating, steam- and gas-power production and for the generation of electric energy. If used in the blast furnace for making iron it gives rise to a number of additional processes, the resulting blast-furnace gas being used for heating, power production and for the generation of electricity.

If the gas be deprived of its more valuable constituents there are obtained, among other things, sulphate of ammonia and sal ammoniac, which is used extensively in the manufacture of explosives and as fertilizer. The yield of ammonium sulphate amounts to about 1.5 per cent. of the weight of coal coked, and its price is about $60 per ton in Germany, or £12 ($60) per ton in England. Germany now controls the world's market with an output of more than 410,000 tons of ammonium salts, representing a selling value of over $25,000,000.

The second important byproduct of coke-oven gas, benzol, represents about 1.5 per cent. of the weight of coal coked. Benzol is used largely in the dye industries, but of late is also being employed largely as fuel for automobiles, its price being about $6 per 100 kg. (220 lb.) or 7½d. (15c.) per gal. in England. The French automobile business absorbs the largest part of the Germany benzol production. Benzol is the mother substance of a large number of aromatic compounds and of artificial dye stuffs; besides, it is used as a solvent and purifier for numerous organic bodies, such as caoutachouc and gums.

The work involved in the recovery of byproducts from coal tar, of which Fig. 2 presents a graphical idea, gives employment to an army of people, from the highly trained scientific expert and investigator, through the ranks of merchants, clerks and skilled laborers, down to the day laborer. The significance of the rational utilization of coal becomes even more apparent if some of the more important tar products are studied in detail. From one-fifth to one-fourth of the coal tar produced is used raw, the rest being subject to development by fractional distillation. The raw tar and such derivates as pitch-tar oil and creosote find extensive use as preservatives of wood and iron from dampness, decay and attacks of insects. Tar paper has now come to be, on account of its simplicity and cheapness, an indispensable building material for roofing and protection. Raw tar is also used to remedy the dust nuisance on roads, and as a binder for crushed stone it gives a smooth, elastic, noiseless, dust-free roadbed. Pitch is extensively used at coal mines as a binder for fine coal and coal dust, permitting these to be compressed into briquettes.

Conclusion

From the foregoing it appears that of all industrial resources coal is the most important material asset of a country. Its rational utilization enables a nation to possess within itself almost all the essentials of national supply, and the possibilities of further development are practically unlimited. To reach this desirable state of commercial independence and technical excellence it is essential to abandon the old wasteful method of treating coal as a fuel and utilizing its heat value only. It is necessary to consider coal as a raw material burdened with latent wealth, and to recover from it, as far as possible, the innumerable byproducts inherent in it. It is further necessary to concentrate the rational utilization of coal at mining centers, to cease transporting it in its crude form, but instead, after separating out its valuable constituents, to transmit it in some superior form of energy, either as gas or electricity, to the markets. By this method it becomes possible to make use of the inferior grades of coal which were hitherto regarded as worthless, at the same time eliminating the transportation factor and insuring maximum industrial efficiency.

How One Mill Beats the Central Station

"What is the situation with regard to the central station?" the editor asked a friend Down East one day. "Is the central station really selling current cheaper than the isolated plant can make power when there is heating to do?"

"Go down to the Lowell Weaving Co. and ask George Dearborn how he came out with it," suggested the friend.

The editor found Mr. Dearborn sitting in a sunny corner of the spick-and-span concrete office building of the weaving company. To my explanation of my interest in his experience with the central-station power, Mr. Dearborn responded, "The electric company here has a young fellow on its staff who can give out the most eloquent and convincing line of argument of anyone who comes in here, and the burden of his song is, 'uninterrupted service and unrivaled economy.' He was so charitably disposed, and so anxious and kind in his disinterested friendship for us that we came near dismantling our plant and hooking up to his feeder, in which case he assured us that our motive power and its worriments would be a thing of the past. We could sell one boiler, run the other at the 'minimum of pressure' (that was another of his pet phrases) and save enough so that we could almost afford to put him on the payroll.

"Well, we had to make a change. We had outgrown our little engine, and to take it out, cut out a bunch of concrete, make the plant over as I wanted to, and put in the bigger engine, would take, as I figured it, eight or ten weeks. So, we got some motors, hooked on to the central station, cast off from our worriments and prepared to glide down this placid stream of 'uninterrupted service and unrivaled economy.'

"We have developed here for the last five years some two or three hundred horsepower. We weigh our coal and take indicator cards four time a day. The coal bills at the end of the year check against the engineer's and fireman's figures and the indicator cards are still on file. Our bills for power averaged in the neighborhood of $26 or $27 a year per horsepower. We were generating power on from 2¾ to 3 lb. of coal per horsepower-hour, and the whole bill for power, including coal, engineer, fireman, oil, waste, repairs, etc., amounted to about $150 a week running under normal conditions. There were some students from the Massachusetts Institute of Technology who came

up here and made a continuous test for a week, and their figures check up with our own.

"Well, sir, there was not a single week out of the ten weeks that the central station was hooked up here that our power bill did not run from $575 to $600, and that's for the power alone. In addition to that we burned a lot of coal to keep the mill warm that would have been included in our little $150 a week when we were running our own plant. So much for the 'unrivaled economy.'

"Now, for the 'uninterrupted service.' We were shut down, three times, in that ten weeks, and I never could tell when we were going to get to going again. Once it was on account of a thunder storm; the other times, something else. We never knew when it was going to happen, and when it did happen, we never knew whether to send the help home or keep them around. If you telephoned, they could not tell you whether the current would be off for two hours or a week. Why, that little Rollins engine that we took out had run five years with only one shutdown to its account, and that was only for about three minutes while they took out a handful of waste that had got into the regulator gears. We have got them cased in on the new one. Come down and look at our little plant. There is not much of it, but it is turning out power about as cheap as any of them."

We went out through the mill and Mr. Dearborn stopped to show us some cotton duck that looked fine enough to make a ball dress of. When I said so, he replied that they had made some out of silk. They make all of the fine duck which is used for sails on the Herreshoff yachts in the international races. They made some which was used on the "Shamrock," too; but that is a secret, for everything on the "Shamrock" was supposed to be British made.

They have a nice, plain, work-a-day plant, with no frills, but with everything excellently adapted to its purpose, convenient to handle and made to give "uninterrupted service," without any spare units, duplicate parts or standby service. The floor of the engine room is a great monolith of concrete, to which is bolted a 24x42-in. Rollins engine running at 93 r.p.m., with 100 lb. initial pressure and a back pressure of from 3 to 4 lb. They use only about one-third of their exhaust steam. The National feed-water heater has ample capacity to heat the feed water nearly to the temperature of the exhaust.

The horizontal return-tubular boilers, of which there are two in place and another being installed, are right alongside the engine, separated only by a heavy wall, and the pipe is so ample and direct that there is little loss of heat or pressure between them. The boilers are 18 ft. by 72 in., with ninety-four 3½-in. tubes. All of the radiating surfaces are heavily covered with magnesia, for that is one of the points upon which Mr. Dearborn places especial stress. "Keep your heat in when you have got it; a little more covering will pay for itself in a little while, and then go on making money for you for years." The boilers are served by a concrete stack 100 ft. in height, with a 72-in. core and a draft, as Mr. Dearborn expresses it "which would snatch the fireman's shovel away if he did not hang on to it."

The engine is belted to a jackshaft with a heavy three-ply belt. If anything went wrong with that belt, or transmission, it would put the mill out of business; but nothing is expected to go wrong. "When I buy a belt," said Mr. Dearborn, "I ask the man the price and then I ask him if I pay him more can he make me a better belt." The belt that was on the engine ran the entire time that engine was in service, without any indication of trouble, and is now running one of the line shafts from the jack. The jackshaft is short and heavy, with ring-oiled bearings, and runs so smooth that, as my host expressed it "You could cool your drinking water on them." He is proud of the fact that the friction load of his mill is only 25 per cent. He uses nothing but turned iron shafting, put together with flange couplings, split wood-rimmed pulleys and ring-oiled hanger bearings. Turned iron shafting, he believes, gives less friction in the bearings and stays straight. The cold-rolled shafting, according to his doctrine, is filled with internal stresses. If you cut a keyway into it and relieve the surface tension, and especially, if you put it up with compression couplings, it will go all out of shape and will "look like a row of cornstalk joints waddling around." His shafting does certainly run so true that it is difficult to see whether it is turning. In the five years' service of the smaller plant, there was only one shutdown on account of the entire transmission. This was caused by a twister cord getting wound around the shaft and fouling one of the rings in one of the hanger boxes, causing it to heat.

The entire cost of the power plant did not exceed $14,500. It will be seen that even when the account is charged with a fair depreciation, interest, taxes, insurance, etc., upon the cost, there is an ample margin between the cost of the power so produced and that at which it could be purchased from the central station.

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Power Distribution for Panama Canal

The electric power distribution for the Panama Canal operation and lighting will require 246 miles of lead-covered cable, which is about 40 per cent. of the total amount placed by the largest electric power company in the United States during the 20 years preceding 1908, states *Engineering News*. These cables will be carried through the lock walls in vitrified clay ducts. On account of the large amount of cable to be placed in the ducts a special device using an electrically operated winch was resorted to.

Before the operation is begun the ducts are carefully cleaned. Then a small steel lead wire is pulled through, and by this means a manila cable is drawn through the duct. The electric cable is attached to the rope, and the winch is set in motion. By this method it is possible to pull 900-ft. lengths of No. 0000 B. & S. gage lead-covered cable at one time, whereas the average length by other methods is about 300 ft. The outside diameter of this cable is a trifle more than 2 in. In such a length the cable passes through two intermediate manholes, and is greased at three points to aid in reducing friction. The advantage of pulling the long sections at one operation is that the making of two joints is saved, and this is a considerable economy. The winch used for the purpose is mounted upon trucks operated over the return tracks of the canal towing system. The cable is placed directly below the floor of the operating tunnel, and, therefore, the pull by the winch is direct.

In addition to the amount of cable mentioned above, the lighting system will require about 112 miles of wire of miscellaneous sizes from Nos. 12 to 00 B. & S. gage.

ELECTRICAL DEPARTMENT

Generator and Prime Mover Capacities*

By D. B. Rushmore and E. A. Lof

The capacity of an alternator is limited not only by the actual load which it is carrying, but also by the power factor. The additional heating at low power factors and constant power is due to the wattless component of the armature current and also to the additional field current necessary to counteract it. In many cases it is possible to predetermine approximately the power factor under which the generators will operate, and a special design to meet this particular condition is largely desirable.

The generator may be designed for unity power factor, while the conditions of operation may give a power factor of 0.8, in which case only a part of the prime-mover capacity can be utilized. In some hydro-electric stations, the waterwheel is either too small or too large for the generator and a like result ensues. Where the generators are installed to handle peak loads, this maladjustment of ratings is a necessity, but with most of the larger systems it is unnecessary to run machines except at or near their maximum ratings. In the past every effort was made to adjust the ratings of generators to the load curves and 25, 50 and 100 per cent. overload guarantees resulted for certain apparatus. Experience and better load conditions have changed this to a maximum rating which has become possible largely through the growth in the size of stations and systems.

The speeds of engine-driven units, both gas and steam, have become nearly standardized, and turbine-driven sets are rapidly approaching the same condition. With waterwheel units, however, the situation is such that the capacity and speed of the generator are often determined largely by the hydraulic conditions and the characteristics of the waterwheel which is to be used.

With steam engines, the point of maximum efficiency is rather marked, and the ratings are usually such that the engine is working under its most economical load at the rating of the electrical generator. With gas engines, however, the efficiency increases with the load beyond the capacity of the engine, and for this reason, the rating of the engine is generally made as nearly as possible to its maximum capacity, leaving a small margin for regulating purposes. With the steam turbine the efficiency curve is usually so flat that it is a question of desirable overload capacity which limits the rating of the turbine. In the waterwheel unit, the efficiency usually falls off rapidly above and below the maximum point, so that the rating of the generator should correspond to the point of maximum efficiency of the waterwheel.

Steam engines and steam turbines are designed to operate with certain variations both in pressure of the steam and conditions of the vacuum. Gas engines must accommodate themselves to variation in the quality of the gas. With the waterwheels, however, most installations are subject to a change in head which varies over a wide range. In many of the low-head installations, the back water may bring about a change in head which is beyond the capacity of the wheel to accommodate, and in some plants additional wheels must be mounted on the same shaft and cut into service at times of low head. One particular instance of this kind is the plant of the Chattanooga and Tennessee River Power Co., where the head may vary from 20 to 42 ft. In most of the large developments, this change in head is the limiting feature in the design of the waterwheel as related to the generator capacity, for in all electrical work it is necessary that the speed of the generator be kept constant.

Waterwheel runners of different designs are compared on the basis of their specific speeds. The specific speed of a waterwheel is the number of revolutions per minute at the point of maximum efficiency that a similar wheel will give when it delivers one horsepower under one meter head. By comparison of the specific speeds, it is possible

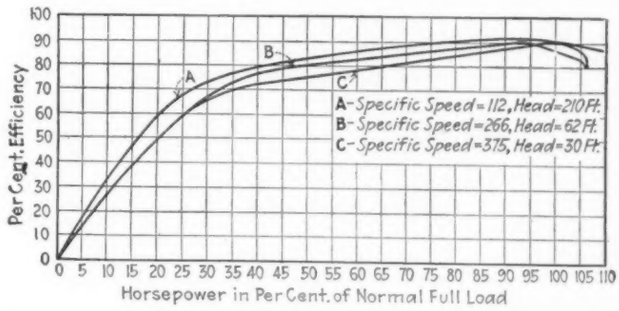


Fig. 1. Variation of Efficiency with Horsepower for Different Specific Speeds

to judge of the characteristics of waterwheel runners without respect to their actual speed, power or head. A high specific speed means a high actual speed and a low specific speed means a low actual speed in revolutions per minute. For this reason, waterwheels with low specific speeds are generally used with high heads in order to make the speed of the generator within the range of good electrical design.

The maximum full-load capacity of a turbine is that point beyond which the output decreases with an increase in gate opening. The margin between the point of maximum efficiency and maximum capacity depends upon the specific speed of the runner and is smaller the higher the specific speed. This is illustrated in Fig. 1, which shows that as the specific speed is increased the point at which maximum efficiency occurs approaches nearer to the power delivered at full gate opening. The specific speed may thus be increased to such an extent that the point of maximum efficiency and maximum output coincide. Therefore, with low heads and high specific speeds it is desirable to operate wheels near their point of maximum output, and to obtain the best results the generator should be designed with consideration given to this point.

*Excerpts from paper presented at midwinter convention of the American Institute of Electrical Engineers.

Referring again to the curves in Fig. 1, it will be noted that the full-load capacity occurs at about 6 per cent. above normal or rated full load in all three cases. This is in accordance with the general practice, the margin being allowed for governing. It is also noted that for curves *B* and *C* the efficiency falls off very rapidly at 6 per cent. overload, and that should the gate be opened still further, the output would reduce instead of increase. If, with wheels of low specific speed, as represented by curve *A*, the gates were still further opened, the power would continue to increase to some extent.

The point of maximum efficiency for wheels represented by curve *A* occurs at about 90 per cent. of normal full load, in the case of *B* at 93.5 per cent., while in the case of *C*, the maximum efficiency occurs just at the point of normal or rated full load. Thus, the power at which the maximum efficiency occurs approaches nearer to full load as the specific speed increases.

With high head wheels, as represented by curve *A*, the efficiency remains high over a large range in power, while for low head wheels, curve *C*, the efficiency falls off rapidly as the power is reduced below the normal full load. For this reason it is desirable to run low head wheels under practically full-load conditions. With high head wheels this is not so important. With wheels, as represented by curve *C*, it is also necessary to allow some margin above the normal full load for governing, as it is desirable to operate the turbine at its point of maximum efficiency. With high head wheels, curve *A*, such a margin need not be allowed.

The curves plotted in Fig. 1 represent operating conditions under constant head. In Fig. 2, is plotted a set of curves illustrating the effect of a varying head. A 10,000-hp. turbine is assumed to operate under a 32-ft. head, the speed being constant for a range of heads from 26 to 38 ft. As the head goes up to 38 ft., the shape of the curve approaches more closely curve *B* in Fig. 1, while, when the head falls to 26 ft., the speed being constant, it approaches more closely to curve *C*. In other words, when operating under a 38-ft. head the speed is lower than the best speed for the runner under that head; while, when operating under the 26-ft. head, the speed of the wheel is higher than the best speed. Under 38-ft. head the point of maximum efficiency is, furthermore, considerably below the normal full load at that head, while under 26-ft. head the power at which maximum efficiency occurs is the actual full load.

In the selection of a speed for any installation, therefore, aside from the cost of the generators, the question of the wheel efficiencies at partial gate openings has considerable bearing. Where a unit is likely to operate under a very wide range in power, it would be advisable to select a unit represented by curve *A*, giving a high efficiency for a considerable range in power.

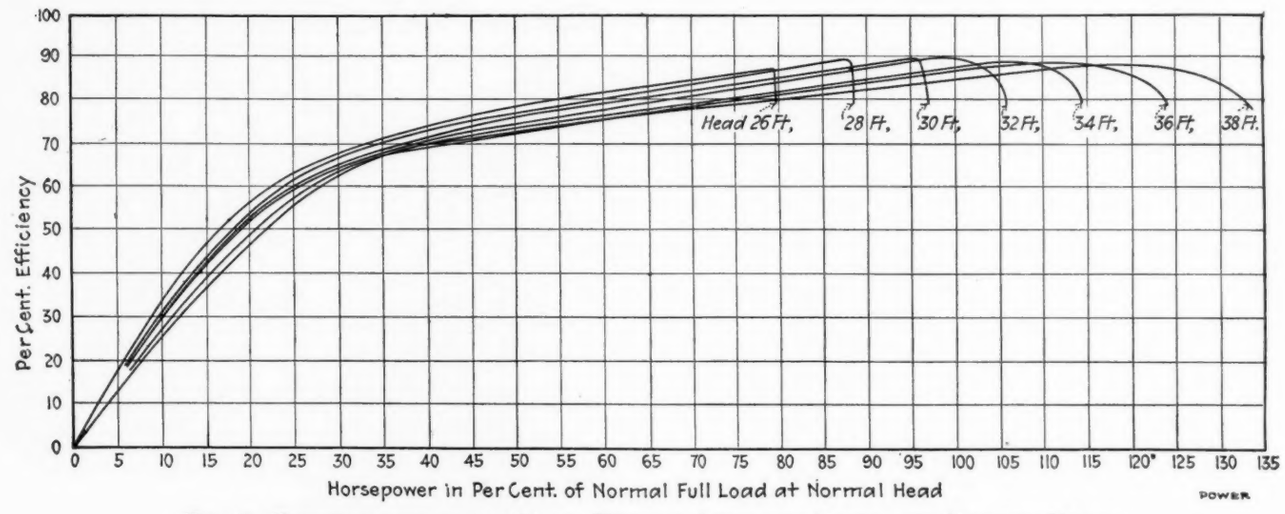
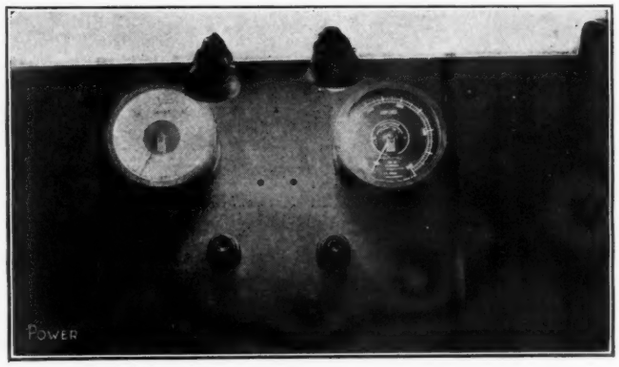


FIG. 2. PERFORMANCE OF 10,000-HP. UNIT AT CONSTANT SPEED AND VARIOUS HEADS

New Departure in Switchboard Meters

One of the elementary principles of scientific illumination is that only those parts should be illuminated that are desired to be seen. This principle has been utilized in a line of 7-in. dial, indicating switchboard meters recently put out by the Westinghouse Electric & Mfg. Co.



BLACK DIAL COMPARED WITH WHITE ONE

A white mark on a black background is much easier seen than a black mark on a white background, and, therefore, these meters are supplied with black dials, the markings on which are white. Their particular field of application is in installations involving poor natural or artificial illumination. The illustration shows the black and white dial meters under identical conditions of illumination, from which it is an easy matter to determine the relative readability of the two types.

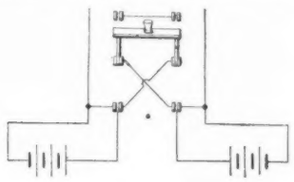
CORRESPONDENCE

Substation Wiring

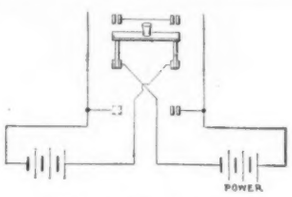
In reading over Thomas G. Thurston's article on "Substation Wiring," in the Mar. 4 issue, I came across an error in Fig. 2. As printed the battery switch *E* is connected up wrong, which shows the two sections of the battery short-circuited at both the lower clips on switch *E*. But the main question still remains, even though switch *E* is connected up properly:

Where did the excess voltage come from, and why did the two signal lamps *R* and *G* burn? The excess voltage over the operating busbar voltage when the signal lights were put on the transformer system was due to the signal light circuit being placed in series with the low-tension side of the 10-kw., 2200- to 110-volt transformer by means of the auxiliary busbar. Such being the case, the transformer acts as a booster, hence adds a proportion of its positive electromotive force to that of the operating busbar.

When the switch *E*, Fig. 2, is closed up, current will flow through the green lamp and give excessive brilliancy, owing to increased electromotive force; thence through the auxiliary busbar and through the low-tension side of the transformer to the negative side of the battery or exciter, whichever is connected to the operating busbars.



As Fig. 2. Shows



As Fig. 2 Should Be

Also current of normal transformer voltage will pass from the negative operating busbar through trip coil *H* and red lamp *R* to the auxiliary busbar and give normal brilliancy.

The reverse action will take place when switch *F* is closed down. Incidentally it may be mentioned that the direct current through the signal light and transformer circuit will be pulsating and unidirectional, and current will flow only when the positive transformer electromotive force is positive with regard to the operating busbars. Therefore, such current will pulsate directly at the rate of 60 per sec.

Ben Dawson.

Cedar Rapids, Iowa.

Commutator Trouble

I have noticed a number of communications in Power on commutator troubles and their causes, but think some of them may have been due to the same cause as some troubles I have experienced.

We have three 110-kw., direct-current generators, direct-connected to three-cylinder vertical gas engines; two of these units gave considerable trouble, the commutators burning in spots and the brushes sparking. All of the usual remedies were tried, but without overcoming the trouble. Some said the trouble was due to soft spots, because the burning always occurred in the same place; others said it was due to the armature winding being connected wrong because the burned spots were at equal distances around the commutator, but experts on armature winding could find nothing wrong. The real cause of the trouble proved to be something entirely different.

In making repairs to one of the engines, it was necessary to dismantle the engine and generator, and when they were being reassembled, I took careful note of every part as it was put in place. When the armature shaft was coupled to the crankshaft, I noticed it did not line up with the field polepieces; this set me to thinking. I found that the other unit was the same—about ⅜ in. out of line. In the third unit, which had never given any trouble, the armature and poles were exactly in line.

The sub-bases of these generators are grouted in cement and it would have been difficult to move them; therefore I removed the field from the sub-base and cut out the bolt-holes to allow bringing the poles in line with the armature, and we have had no more trouble.

The engine cranks are set at 120 deg., and each of the burned spots was in the same relation to each of the crankpins. The connecting-rods had a tendency to hold the crank in one position, but the field was creating a magnetic pull against the connecting-rods and caused the brushes to spark at this point on each revolution of the commutator and this continuous sparking caused the burned spots.

I find it a good policy to wipe commutators off several times a day with a clean cloth, and to go over all brushes quite often, seeing that they work freely in the holders, that the contact points are free from copper, and that all other conditions are right.

H. H. Delbert.

Titusville, Penn.

Brushes Boiled in Mineral Oil

We have several direct-current generators and motors and have experienced the usual troubles with them; one 25-kw. machine in particular giving considerable trouble on account of sparking and roughing the commutator on heavy loads. Consequently, we had to sandpaper and turn the commutator very frequently.

About a year ago I read in Power about boiling the brushes in mineral oil. At first I hesitated to try it, as I thought it would gum up under the brushes; but having a spare set I finally tried it, and found that it worked to perfection. Whereas, the brushes had previously been emitting the usual destructive blue sparks, this changed to a thin streak of a dull red which followed around the commutator a short distance from the brush. After wiping off the commutator frequently for a day it ran cool without a spark. I now change and boil the brushes every two or three months, and have good commutation on this machine.

The effect on the brush by boiling is to soften it, and on a slightly rough commutator the rough places become filled with the carbon from the brushes; this carbon adheres and improves commutation, thus preventing sparking. Where there were several bars blackened and rather low, after a time they became polished and now as soon as the commutator gets the least bit rough I always boil the brushes and it effects a cure.

Brockville, Ont., Can. S. G. Rose.

GAS POWER DEPARTMENT

Heat Balance of a Gas Engine*

By Julian C. Smallwood

A complete economy test of a gas engine should include, besides the fuel consumption, the determination of the various losses so that their distribution may be studied. All of the heat energy of the fuel supplied must be accounted for in the form of useful work and losses. The equation between these quantities is called the heat balance.

The losses involved in the operation of a gas engine are mechanical friction, heat carried away by the jacket water, by the dry exhaust gases (sensible heat), by steam in the exhaust, by unburned gas, and stray losses, such as radiation, etc. It is convenient and logical to base the calculation of these losses upon one cubic foot of fuel gas under standard conditions of temperature and pressure. The heat balance may then be written, "Heat supplied by 1 cu.ft. of gas equals the heat from 1 cu.ft. turned into useful work plus the heat lost to friction, etc."

To measure the useful work, friction and jacket losses on this basis, it is necessary to meter the fuel supplied in a given time. The other quantities of the heat balance require the analysis of both the fuel and the exhaust gases. Generally, for the fuel analysis, a chemist's services are needed. Strictly speaking, a fair sample of the gas used during the test should be analyzed, but when the fuel is such as city illuminating gas which does not vary much from day to day, an average analysis is fairly representative and may be used without much error.

TABLE I. PRINCIPAL CONSTITUENTS, IN PERCENTAGE OF VOLUME, OF GAS-ENGINE FUELS

"Low" means less than one per cent.

	Producer Gas, Per cent.	Illuminating Gas, Per cent.	Natural Gas, Per cent.
Carbon monoxide, CO	25	15	low
Hydrogen, H_2	12	45	2
Methane, CH_4	2	25	95
Ethylene, C_2H_4 / Benzol, C_6H_6 / Heavy hydrocarbons	low	5	low
Oxygen, O_2	low	1	low
Sulphuretted hydrogen, H_2S / Sulphur dioxide, SO_2	low	low	low
Carbon dioxide, CO_2	5	4	low
Nitrogen, N_2	55	5	2
Higher heating value, B.t.u. per cu.ft.	150	650	1050

Table 1 gives the average composition of the principal gas-engine fuels. The analysis of the exhaust gases may be made with an Orsat apparatus. The chief constituents are CO_2, O_2 and N_2, but an analysis for CO should not be omitted. A determination of H_2 is desirable, but is not necessary if the CO is less than 1 per cent., as it is in proper operation. Since the CO is the least readily burned of the gas constituents, it is a fair assumption that the hydrogen and hydrocarbons are completely burned if the CO is nearly all burned, and experiment bears out this assumption.

Knowing the gas analysis, the weight of gases in the exhaust pipe per cubic foot of fuel supplied may be calculated from the formulas herein given. These are deduced upon the hypothesis that all the carbon in the fuel appears in the exhaust-gas analysis and that none other does. Contrary to this is the fact that some carbon is left as a deposit in the cylinder, but it is so small compared with the total carbon used as to be negligible. Also, if the lubricating oil burns, it will appear as CO_2 in the exhaust, so care should be taken that this condition does not obtain.

Other quantities, useful for purposes of comparison, that may be figured from the analyses are the number of cubic feet of air consumed per cubic foot of fuel and the "excess coefficient." The latter is the ratio of the former quantity to that required for complete theoretical combustion.

Sampling

If the fuel gas actually used on the test is analyzed, it is best to take a continuous sample covering the whole time of the test. This can be done by connecting a 5- or 3-gal. flask with rubber tubing to the gas main and siphoning water from the flask at a slow and uniform rate, thus drawing in the sample. The exhaust gas may be sampled the same way, but, if preferred, a number of samples covering equal lengths of time and taken at equal time intervals may be used and the results of their separate analyses averaged. A convenient method is to collect one sample while analyzing another, thus making a continuous operation. This has the advantage of showing the uniformity of combustion as the test proceeds. Snap samples should not be taken.

Duration of Test

This should last several hours, preferably, but, if on a small engine, a test of only one hour will give valid results, if the engine is previously run for 15 or 20 min. under the conditions to be maintained during the test. The readings to be mentioned later should be taken sufficiently close together to obtain a fair average. The useful horsepower should be maintained constant and other conditions kept as uniform as possible.

To show how a heat balance is derived the data from a gas-engine test will be worked through for the various results. Table 2 gives these data, and also the notation used in the formulas. The fuel used was illuminating gas manufactured in Syracuse, an average analysis of which is given in columns (1) and (2) of Table 3.

Fuel Consumption

Since the weight of fuel in 1 cu.ft. varies with the pressure and temperature, the gas-meter readings should be referred to standard conditions, namely, a pressure of 29.92 in. of mercury and a temperature of 32 deg. F. The following formula may be used. (See Table 2 for notation.)

$$V' = 16.4 \times \frac{B + \frac{P}{13.6}}{T_f + 460} \times V$$

*Copyright, 1913, by Julian C. Smallwood, associate professor of experimental engineering, Syracuse University.

The barometer reading may be obtained in most cases from the nearest weather bureau; whereas, P and Tf necessitate the use of a water manometer and a thermometer at the point in the gas main where the gas meter is placed. For small engines, the usual gas meter may be used, or a gasometer, but for large sizes meters of the pitot, venturi or orifice type may be used. It is assumed that the reader is familiar with the methods of measuring brake and indicated horsepowers. When the engine is so large as to be inconvenient to brake, the useful horsepower may be approximated by subtracting from the indicated horsepower at the running load its value when the engine is running free.

TABLE 2. DATA FROM A GAS-ENGINE TEST, AND NOTATION USED IN FORMULAS

B.hp. = Brake horsepower	= 25
I.hp. = Net indicated horsepower	= 30.1
V = Cubic feet of fuel gas per hour by meter	= 414
V′ = Corrected to 29.92 inches of mercury and 32° F.	= 29.5
B = Barometer reading, inches of mercury	
P = Pressure of fuel gas above atmosphere, at meter, inches of water	= 4
F = Fuel consumption, standard cubic feet of gas per b.hp.-hr.	
Tf = Temperature of fuel gas at meter deg. F.	= 68
tj = Temperature of ingoing jacket water, deg. F.	= 61
Tj = Temperature of outgoing jacket water, deg. F.	= 133
Te = Temperature of exhaust gases, deg. F.	= 750
t = Temperature of air near engine, deg. F.	= 70
Wj = Weight of jacket water, pounds per hour	= 1210
Wd = Weight of dry exhaust gas from the combustion of 1 cu.ft. of fuel, lb. per standard cu.ft.	
Wv = Weight of water vapor from the combustion of 1 cu.ft. of fuel, lb. per standard cu.ft.	
R = Ratio by volume of air consumed to fuel gas	
r = Ratio required for theoretical combustion	
X = Excess coefficient	
c, h, g = Sums of columns (3), (4), (5), Table 3, respectively.	
Volumetric Analysis of Exhaust Gas, Per Cents.	
D = Carbon dioxide, CO_2	= 9.0
O = Oxygen, O_2	= 8.8
M = Carbon monoxide, CO	= 0.5
H = Hydrogen, H_2	Not Analyzed
N = Nitrogen, N_2	= 81.7

Applying the data of Table 2,

$$V' = 16.4 \times \frac{29.5 + \frac{4}{13.6}}{68 + 460} \times 414 = 385 \; cu.ft. \; per \; hr.$$

From which the fuel consumption per brake horsepower-hour is

$$F = \frac{385}{25} = 15.4 \; cu.ft.$$

Heat Supplied

It is preferable to use a calorimeter to determine the heat in one cubic foot of the fuel gas, but if one is not available a working value may be obtained by calculation from the fuel-gas analysis.

The gas-engine code of the A. S. M. E. specifies that the "higher" heat value be used; that is, the heat obtainable from the fuel, including the latent heat of vaporization of the water formed by combustion of the hydrogen. Some authorities contend that the lower heat value is more logical to use in this connection since the gas engine cannot avail itself of this latent heat, but for the sake of uniformity the recommendations of the code will be followed.

TABLE 3. ANALYSIS OF FUEL GAS USED DURING TEST, AND COMPUTATION OF QUANTITIES USED IN FIGURING RESULTS

(1) Constituents of Fuel	(2) Percent. by Volume	(3) Subscript of C×(2)	(4) Subscript of H×(2)	(5) Subscript of O×(2)	(6) Higher Heat Value of (1)	(7) (6)x(2) / 100
CO	23.5	23.5	....	23.5	342	80.5
H_2	36.4	....	72.8	....	348	127
CH_4	15.3	15.3	61.2	....	1065	163
C_2H_4	6.9	13.8	27.6	....	1680	116
C_6H_6	3.9	23.4	23.4	....	4000	156
O_2	1.4	....	....	2.8	....	.....
CO_2	4.2	4.2	....	8.4	....	.....
N_2	8.4	....	....	....	....	.....
Total..		80.2 =c	185.0 =h	34.7 =g		643 = higher heat value of fuel

It is preferable to use a calorimeter to determine the heat in one cubic foot of the fuel gas, but if one is not available a working value may be obtained by calculation from the fuel-gas analysis.

The gas-engine code of the A. S. M. E. specifies that the "higher" heat value be used; that is, the heat obtainable from the fuel, including the latent heat of vaporization of the water formed by combustion of the hydrogen. Some authorities contend that the lower heat value is more logical to use in this connection since the gas engine cannot avail itself of this latent heat, but for the sake of uniformity the recommendations of the code will be followed.

To calculate the higher heat value of the fuel, multiply the higher heat value of each of the constituent gases, in B.t.u. per standard cubic foot, by its volume percentage, as shown by the fuel-gas analysis, and divide by 100. Add these products together: the sum is the heat value of the fuel gas. In Table 3, Column (6) gives the heat values of the combustible constituents. Column (7) gives the heat supplied by the amount of each constituent in 1 cu.ft. of the fuel gas used in the test under consideration. The sum of these, or the heat value of this fuel, is 643 B.t.u. per standard cubic foot.

Heat Converted into Useful Work

The heat equivalent of 1 b.hp.-hr. is 2545 B.t.u. Since the fuel consumed to produce this power is F cu.ft., the heat equivalent of the useful power per cu.ft. of fuel is

$$\frac{2545}{F}$$

Applying the data of the example this is

$$\frac{2545}{15.4} = 165 \; B.t.u.$$

Dividing this by the heat value gives

$$\frac{165}{643} = 0.256 \; or \; 25.6 \; per \; cent.$$

This is the thermal efficiency.

Heat Lost to Mechanical Friction

Friction horsepower = Indicated horsepower — brake horsepower

To base this upon the heat of 1 cu.ft. of fuel, a formula derived the same as the one last given is used, namely,

$$Friction \; loss, \; B.t.u. = \frac{2545 \times friction \; horsepower}{V'}$$

Applying the data of the example, this is

$$\frac{2545 \; (30.1 - 25)}{385} = 33.7 \; B.t.u.$$

or 33.7 ÷ 645 = 5.2 per cent.

Heat Lost to Jacket Water

This is the heat absorbed in a given time divided by the fuel used in the same time. Considering hourly quantities,

$$Jacket \; loss = \frac{W_j \; (T_j - t_j)}{V'}$$

The weight W_j may be measured by a barrel or tank on a platform scales. The temperatures should be read with two thermometers, one in the discharge tank (provided the piping is arranged to discharge without material radiation loss); the other in a thermometer well in the inlet pipe or in a receptacle drawing water from a tap in the main situated similarly to the jacket as to temperature.

Applying the data of the example, the

$$Jacket \; loss = \frac{1210 \; (133° - 61°)}{385} = 226 \; B.t.u.$$

or 226 ÷ 643 = 35.2 per cent.

Loss of Sensible Heat to Dry Exhaust Gases

This determination involves the measurement of the temperature rise of the fuel and air mixing with it, and of the weight of dry exhaust gases. For the specific heat, an average value, 0.245, may be used with sufficient accuracy. Then

Loss to dry exhaust gas $= 0.245 \times Wd \times (T_e - t)$

Wd being the weight per cubic foot of fuel, obtained as follows:

$$Wd = \frac{c\,\{11\,D + 8\,O + 7\,(M + N)\}}{9000\,(D + M)}$$

The value c is obtained as shown by Table 3. The subscript of the C in each fuel gas constituent shown in Column (1) is multiplied by the volume percentage of that constituent as shown in Column (2). Column (3) shows these products, the sum of which is the value of c.

The weight formula is based on the assumption that the hydrogen and hydrocarbons are completely burned. Hydrogen in the exhaust, not analyzed for, generally produces but little effect on the weight of the exhaust gas since it is small in volume and density.

The temperature in the exhaust pipe should be taken by a high-reading thermometer (1000 deg. F.) inserted in a well filled with fine, dry sand, located as near the engine as possible.

Applying the data of the example,

$$Wd = \frac{80.2\,\{(11 \times 9 + 8 \times 8.8 + 7\,(0.5 + 81.7)\}}{9000\,(9 + 0.5)}$$

$$= 0.703\ lb.\ per\ cu.ft.\ of\ fuel$$

and the loss to the dry exhaust gas equals

$$0.245 \times 0.703 \times (750 - 70) = 118\ B.t.u.$$

or $118 \div 643 = 18.3$ percent.

Loss to Water Vapor in Exhaust

Since the higher heat value of the gas is used, the latent heat of the water vapor should be included. It is then necessary to find the total heat of the steam in the exhaust gas above the heat of the liquid at room temperature. If the pressure of the steam were known, its total heat could be found by reference to the steam tables; but since the steam is only a small part of the exhaust gas, its pressure is very much less than that of the mixture and is inconvenient to figure. Professor Diederichs has proposed the following expression for total heat under these conditions which agrees substantially with the curve furnished by Professor Marks.

Heat loss to water vapor $= W_v\,(1090 + 0.46\,T_e - t)$

W_v, the weight of water vapor from 1 cu.ft. of fuel, may be figured from

$$W_v = 0.000251\left(h - \frac{2\,Hc}{D + M}\right)$$

The value h is found the same as c, as shown in Table 3. If hydrogen in the exhaust has not been analyzed for, the term $2\,Hc \div (D + M)$ may be neglected. The formula ignores the small amount of water vapor brought in with the air and with the fuel gas.

Applying the data of the example,

$W_v = 0.000251 \times 185 = 0.0464\ lb.\ per\ cu.ft.\ of\ fuel$

and

Loss to water vapor $= 0.0464\,(1090 + 0.46 \times 750 - 70) = 64.3\ B.t.u.$

or $64.3 \div 643 = 10$ per cent.

Heat Lost by Unburned Fuel Gas

Considering only carbon monoxide and hydrogen in the exhaust, this equals

$$\frac{3.45\,c\,(M + H)}{D + M}$$

If hydrogen has not been analyzed for, omit H from this expression.

Applying the data of the example,

$$Loss\ by\ unburned\ fuel = \frac{3.45 \times 80.2 \times 0.5}{9 + 0.5}$$

$$= 14.65\ B.t.u.$$

or $14.65 \div 643 = 2.3$ per cent.

Heat Lost to Radiation, Etc.

This is found by subtracting from the heat value of the fuel the other quantities of the heat balance, as just found, or by subtracting from 100 per cent. these quantities expressed in percentages. The "unaccounted for" heat includes that used for pumping the fuel mixture into the cylinder if the area of the lower loop has been subtracted from the upper loop of the indicator diagrams. This quantity could be separately accounted for, but it is generally sufficient to include it in the friction loss as is done when the lower loop is altogether ignored, or to list it under radiation.

Applying the data of the example,

Loss to radiation, etc., $= 643 - 165 - 33.7 - 226 - 118 - 64.3 - 14.6 = 21.4\ B.t.u.$

or $21.4 \div 643 = 3.4$ per cent.

Cubic feet of air per cubic foot of gas. This may be figured from the following:

$$R = \frac{c}{21}\left\{\frac{D + O + 0.5\,(M + H)}{D + M} + \frac{h - 2\,g}{4\,c}\right\}$$

If hydrogen in the exhaust has not been analyzed for, omit H from the foregoing.

Applying the data of the example,

$$R = \frac{80.2}{21}\left\{\frac{9 + 8.8 + 0.5 \times 0.5}{9 + 0.5} + \frac{185 - 2 \times 34.7}{4 \times 80.2}\right\}$$

$$= 8.63\ cu.ft.$$

Excess Coefficient

The air required for the complete theoretical combustion must first be figured. This is

$$r = \frac{2\,c + 0.5\,h - g}{42}$$

Applying the data of the example,

$$r = \frac{2 \times 80.2 + 0.5 \times 185 - 34.7}{42} = 5.2\ cu.ft.$$

from which

$$X = \frac{R}{r} = \frac{8.63}{5.2} = 1.66$$

HEATING AND VENTILATION

Pipe Sizes for Forced Hot Water Heating

By W. L. Durand

In the design of forced hot-water systems, the most common method of determining pipe sizes is on the basis of some allowable velocity. A 20-deg. F. drop in the radiator is assumed and the total heat required divided by 20 will give the pounds of water required. Water at 180 deg. F. weighs 60.5 lb. per cu.ft. Hence, the pounds of water required divided by this figure will give the cubic feet per hour. This divided by the velocity will give the cross-sectional area of the pipe. Thus

$$\frac{\textit{Pounds of water per hour}}{60.5 \times 3600 \times \textit{allowable vel. in ft. per sec.}} = \textit{area of pipe}$$

This velocity is ordinarily made from 2 or 3 ft. per sec. to 8 or 10, the smaller values being used for branch connections and risers where they serve to compensate the rapid increase in frictional resistance with decrease in diameter of pipe. Then the friction head is computed on the basis of the layout determined by the above method, and, if it falls within the limits set, well and good. If not, it is necessary to use some other velocity and recompute the pipe sizes.

This method is long and does not give results in keeping with good engineering and, due to the rapid change in friction with varying pipe sizes, does not give a balanced system.

What would appear to be a more logical method of determining pipe sizes would be to fix the total friction head per 100 ft. and from the total length of the system determine the allowable friction head per 100 ft. If a radiation table is compiled between the size of pipe and friction head, it would be a simple matter through its use to determine the pipe sizes for any system. Then, the final result would be a definite friction head and a constant drop throughout the whole system and not a small drop in one part and a large drop in another, as with the method outlined above.

The accompanying table gives the velocity per second, the cubic feet of water per minute and the radiation based on 170 B.t.u. per sq.ft. per hr. and a 20-deg. F. drop for pipes from 1 to 10 in. and friction heads per 100 ft. from 1 to 5 ft. This table could be extended if the conditions required; but it will be found to cover a large part of forced hot-water work.

VELOCITY AND VOLUME OF WATER AND RADIATION FOR DIFFERENT FRICTION HEADS

Friction Head per 100 Ft. Length of Pipe in Feet

Size Pipe, in.		1	1¼	1½	2	2½	3	3½	4	4½	5
	Vel.	1.21	1.36	1.50	1.75	1.96	2.16	2.34	2.51	2.68	2.83
1	Cu.ft.	0.436	0.490	0.540	0.630	0.695	0.777	0.841	0.904	0.965	1.02
	Rad.	186	210	230	270	297	332	360	386	412	435
	Vel.	1.50	1.68	1.85	2.14	2.41	2.65	2.88	3.09	3.29	3.47
1¼	Cu.ft.	0.936	1.05	1.15	1.33	1.50	1.65	1.80	1.93	2.05	2.16
	Rad.	400	450	490	570	640	705	770	825	875	920
	Vel.	1.60	1.80	2.00	2.31	2.61	2.88	3.11	3.34	3.58	3.76
1½	Cu.ft.	1.35	1.52	1.69	1.95	2.21	2.44	2.63	2.82	3.03	3.18
	Rad.	575	650	720	830	945	1040	1125	1200	1300	1360
	Vel.	1.90	2.13	2.35	2.73	3.08	3.40	3.69	3.93	4.20	4.44
2	Cu.ft.	2.81	3.15	3.48	4.04	4.56	5.03	5.46	5.82	6.22	6.57
	Rad.	1200	1350	1490	1720	1950	2150	2330	2480	2660	2810
	Vel.	2.14	2.40	2.64	3.08	3.48	3.81	4.15	4.45	4.73	5.00
2½	Cu.ft.	4.28	4.80	5.28	6.16	6.96	7.62	8.30	8.90	9.46	10.00
	Rad.	1830	2050	2260	2630	2980	3260	3540	3800	4050	4270
	Vel.	2.44	2.75	3.02	3.53	3.98	4.40	4.76	5.10	5.40	5.72
3	Cu.ft.	7.51	8.47	9.30	10.9	12.3	13.5	14.7	15.7	16.6	17.6
	Rad.	3210	3620	3980	4650	5250	5760	6270	6700	7100	7500
	Vel.	2.67	3.00	3.31	3.86	4.33	4.80	5.20	5.56	5.90	6.25
3½	Cu.ft.	11.0	12.4	13.6	15.9	17.8	19.8	21.4	23.0	24.4	25.8
	Rad.	4700	5300	5800	6800	7600	8450	9150	9800	10,400	11,000
	Vel.	2.89	3.25	3.58	4.18	4.70	5.15	5.60	6.00	6.40	6.75
4	Cu.ft.	15.3	17.2	19.0	22.2	24.9	27.3	29.7	31.8	33.9	35.8
	Rad.	6530	7350	8100	9500	10,600	11,650	12,700	13,600	14,500	15,600
	Vel.	3.08	3.47	3.80	4.42	5.00	5.50	5.96	6.40	6.80	7.20
4½	Cu.ft.	20.4	23.1	25.2	29.4	33.2	36.6	39.6	42.5	45.2	48.0
	Rad.	8700	9900	10,800	12,600	14,200	15,600	16,900	18,150	19,300	20,500
	Vel.	3.80	3.72	4.10	4.80	5.40	5.93	6.45	6.90	7.35	7.75
5	Cu.ft.	27.5	31.0	34.2	40.0	45.0	49.4	53.7	57.5	61.1	64.5
	Rad.	11,750	13,250	14,600	17,100	19,200	21,100	23,000	24,600	26,100	27,500
	Vel.	3.75	4.22	4.67	5.40	6.10	6.70	7.25	7.80	8.30	8.75
6	Cu.ft.	45.1	50.8	56.2	65.0	73.5	80.6	87.3	94.0	100	105
	Rad.	19,300	21,700	24,000	27,800	31,400	34,500	37,200	40,100	42,700	45,000
	Vel.	4.10	4.61	5.08	5.90	6.65	7.30	7.90	8.45	9.00	9.50
7	Cu.ft.	66.2	74.5	82.0	95.3	107	118	127	136	145	153
	Rad.	28,200	31,800	35,000	40,700	45,750	50,500	54,300	58,200	62,000	65,500
	Vel.	4.38	4.92	5.43	6.35	7.15	7.85	8.50	9.15	9.75	10.4
8	Cu.ft.	91.5	103	113	133	149	164	177	191	204	217
	Rad.	39,000	44,000	48,300	57,000	63,700	70,000	75,700	81,600	87,100	93,000
	Vel.	5.02	5.68	6.25	7.25	8.20	9.00	9.75	10.5	11.1	11.8
10	Cu.ft.	165	187	206	238	270	296	320	345	365	388
	Rad.	70,400	80,000	88,000	101,500	115,200	126,000	137,000	147,000	156,000	166,000

The writer has not as yet in operation a forced hot-water system based on the above table, as it was only recently worked up as a possible improvement over the usual method. Any criticism of this method would be gladly welcomed.

A Mixed Heating System

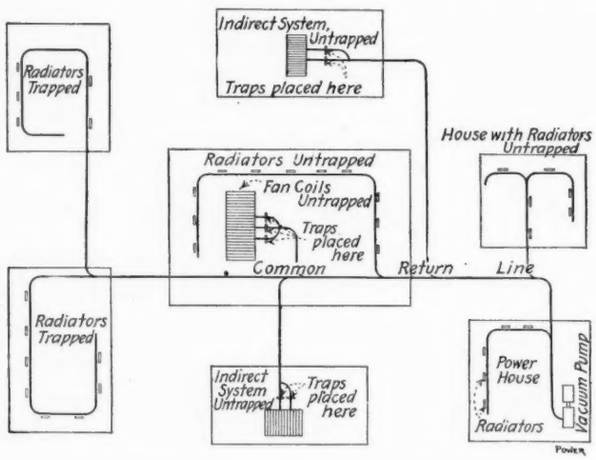
By Edward T. Binns

Some changes tending toward economy in the heating and power plant of a large philanthropic institution may

prove of interest. The plant was out of repair and much loss had been occasioned by the violation of certain basic rules at the time of installation.

One large administration building and a number of smaller buildings are heated and lighted from a central power plant. In the main building the indirect system of heating is employed for the three upper floors. An engine-driven fan, 3½x7 ft., delivers fresh air over some 8000 ft. of 1-in. pipe. These coils are heated jointly by the exhaust from the fan engine and a branch from the exhaust line which extends to all parts of the system. Two other houses are heated by indirect radiation, having engine-driven fans of proportionate size. Four other houses are heated directly and have radiators throughout. One vacuum pump must handle the water of condensation from all. The return ends of the radiators in two of the cottages are equipped with thermostatic traps. None of the drain lines from the fan coils were trapped, one being 2 in. in diameter and two 1¼ in. There were no traps on about 35 radiators in the main building and power house.

All returns from trapped and untrapped units were run to the vacuum pump by way of a common return line. As a result of this combination no end of trouble was experienced. Nothing could prevent steam from com-



Connection of Return Lines

ing back through the return line to the pump and destroying the vacuum. Besides, the various lines and heating units, being but poorly drained, gave continual trouble from leakage. Much trouble was also experienced from coils and radiators being short-circuited.

As an only recourse for producing vacuum sufficient to keep the system water free, the cold-water spray valve at the pump had been perpetually kept open. As the vacuum pump discharged into an open feed-water heater, this caused it to overflow constantly and waste a great deal of hot water during the season.

To relieve the trouble, nonvapor traps were installed at the points where the open steam lines which drained the banks of coils entered the common return. A trap of the thermostatic type was selected, and in ½-in. sizes. Water, but no steam, was conveyed to the return line, and the pump worked successfully. Radiators and coils were kept more free of water by the maintenance of a better vacuum, and the firemen had no more trouble in keeping up steam.

Another source of waste was the carrying of an uncertain and often excessive pressure in the exhaust line, and besides the waste from too much back pressure upon the engines, a considerable amount of steam and hot water was often ejected from the vent pipe of the feed-water heater. It might be mentioned that there was no feed-water thermometer, and that the back-pressure gage was not working. These two valuable adjuncts were speedily fitted up. Exhaust steam filled but a fractional part of the requirements, and it was necessary to use considerable live steam admitted through a reducing valve.

By replacing leaky blowoffs, broken globe-valve seats, worn and broken piston rings, etc., both the saving and satisfaction promise to be of substantial character.

Pressure Losses in Piping System*

The losses in the piping system are made up of two parts, the dynamic losses and the friction losses. The dynamic losses are due to changes either in the direction or velocity of the air flow and are composed of the loss of entrance and of the loss in elbows and connections. The first is the pressure required to produce velocity itself in the pipe and may vary anywhere from 1 to 1.5 times the velocity head, i.e., the pressure corresponding to the velocity, depending on whether the pipe is connected directly to the fan outlet or through the plenum chamber. It is expressed as a multiple of the pressure corresponding to the average velocity produced in the pipe. Where the velocity in the pipe is the same as at the fan outlet, this may still be considered a loss, in view of the fact that with a reduction of velocity through a gradually diverging outlet to a larger area, the difference between the velocity head at the fan and the velocity head in the pipe is largely utilized by conversion to static pressure.

The other chief source of dynamic loss is in the elbows and depends on the radius of curvature of the elbow and not on the size of the elbow or the velocity of the air. This loss may be expressed directly in per cent. of the velocity head, and, with a round, five-piece elbow, having a center line radius of one diameter, the loss will be 25 per cent. of the average velocity head. With a five-piece elbow, having a center line radius of 1½ diameters, the loss will be 17 per cent., or only two-thirds that of the first one.

The second source of pressure loss in the piping system is due to the friction of the air against the sides of the pipe. This loss will vary directly as the length of the pipe, or as the square of the velocity, and inversely as the diameter of the pipe. As the length is a fixed quantity for any system, the only factors subject to modification are the diameter and the velocity, which determine the relation between the power cost and the piping cost.

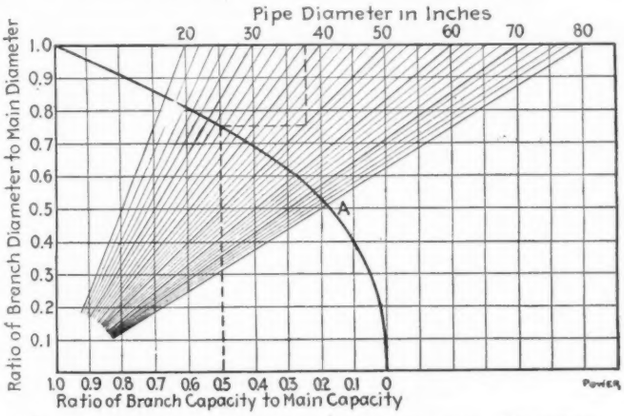
As in the case with the heaters, it is the usual engineering practice to proportion the piping arbitrarily, either from assumed velocities depending on the velocity of the air at the fan outlet, or, in better engineering practice, by determining the velocity which will give an assumed resistance considered suitable and within the fan capacity. It is the usual practice among fan manufacturers gradually to increase the velocity of the air in

*Latter part of paper on "Operating Cost of Indirect Heating Systems," by Frank L. Busey and Willis H. Carrier, read before the American Society of Heating and Ventilating Engineers. First part of paper appeared Mar. 18, 1913.

the main conduit, as the latter is decreased in size, owing to partial delivery of the air through branch outlets.

From the accompanying diagram, the sizes of the branches or reduced piping may be proportioned directly from the assumed size of the main pipe.

For example, assume a branch, intended to carry 50 per cent. of the capacity of the main pipe, which is 50 in. in diameter. Starting at the bottom of the chart with 50 per cent. capacity, pass upward until line *A* is intersected; thence, horizontally, to the diagonal line marked 50, and upward to the scale at the top of the chart, which gives a diameter of 38 in. for the branch


Pipe Diameter in Inches
20 30 40 50 60 70 80
Ratio of Branch Diameter to Main Diameter
1.0 0.9 0.8 0.7 0.6 0.5 0.4 0.3 0.2 0.1
A
1.0 0.9 0.8 0.7 0.6 0.5 0.4 0.3 0.2 0.1 0
Ratio of Branch Capacity to Main Capacity
POWER

PROPORTIONING PIPING TO ALLOW FOR FRICTION

pipe. This is so proportioned as to give the same friction per foot of length as in the main pipe. This permits the resistance of the entire system to be calculated as though it was of one diameter, with a corresponding velocity throughout its entire length.

VELOCITY FOR MAXIMUM ECONOMY

A decrease in velocity increases the size and cost of the air conduit, but decreases the cost of power consumed in overcoming the conduit or piping resistance. From a point of economy, the question to be determined is, what relation between power cost and conduit cost, as determined by the velocity, will give the minimum annual total cost. This relationship is shown in the appendix to be

$$\left(\frac{V_m}{V_o}\right)^3 = 0.335\left(\frac{C_{wo}}{C_{po}}\right) \qquad (1)$$

$$V_m = 0.7\,V_o\left(\frac{C_{wo}}{C_{po}}\right)^{\frac{1}{3}} \qquad (2)$$

Where C_{po} and C_{wo} represent, respectively, the cost of power to overcome piping resistance and the annual allowance on the cost of the piping for interest and depreciation at an assumed velocity V_o, and V_m is the relative velocity required for maximum economy.

Comparing these relationships with those obtained for the heater, it is evident that they are almost identical. It may be seen, in this case, that, for the maximum economy, the annual cost of power consumed for piping resistance should be practically one-third of the annual interest and depreciation charges, based on the initial cost of piping.

PRACTICAL APPLICATIONS

For the purpose of illustrating the application of the foregoing principles to a system of galvanized-iron piping, different cases will be assumed and the results shown. A system handling 30,000 cu.ft. per min. at a velocity of 1950 ft. per min. will require a pipe 53 in. in diameter or an area of 15.32 sq.ft. These quantities will be taken as a constant condition, but different arrangements considered in the system of piping.

Assuming one straight duct, 200 ft. long and 53 in. in diameter, delivering all of the air at the end farthest from the fan, there will be two sources of loss to be overcome by the fan. First, the dynamic loss due to the velocity of 1950 ft. per min. (or one velocity head), and second, the loss due to friction, amounting to one velocity head in each forty diameters of length. The pressure due to the velocity of 1950 ft. per min. in the pipe, will be 0.237 in. water gage. The loss of pressure due to friction will be

$$\frac{200}{4.42 \times 40} = 1.13\ \textit{velocity head}$$

This loss expressed in inches of water will be

$$0.237 \times 1.13 = 0.257\ \textit{in.}$$

and the total loss will be the sum of these two, or 0.494 in. In the piping system, a part of the velocity is converted to static pressure, hence the power calculation should be based on total pressures with a corresponding fan efficiency of 50 per cent. At a rate of $20 per hp.-yr., the annual cost due to the piping resistance will be

$$C_{po} = 30{,}000 \times 0.000324 \times 0.494 \times 20 = \$96$$

A round, galvanized-iron pipe, 53 in. in diameter, would be made of No. 18 iron, weighing 2.3 lb. per sq.ft., and would contain 15.2 sq.ft. per running foot. This would make 32.7 lb. per running foot, or a total of 6540 lb. for the entire pipe. Allowing 25 per cent. annually for interest and depreciation on an initial cost of, say, 10c. per lb., the yearly allowance would be 2.5c. per lb. of iron. Then, the yearly allowance for interest and depreciation would be

$$C_{wo} = 6540 \times 0.025 = \$163.50$$

From equation (2) it may be determined that for the most economical conditions the velocity of the air in the pipe should be

$$V_m = 0.7 \times 1950\left(\frac{163.5}{98.0}\right)^{\frac{1}{3}} = 1620\ \textit{ft. per min.}$$

Assuming the case where the 30,000 cu.ft. per min. is to be uniformly distributed through a galvanized-iron pipe 200 ft. long, with equal openings every 20 ft. of its length, each discharging 3000 cu.ft. per min., will give an example of another common form of installation. Referring to the chart, it may be seen that, if the first 20 ft. of pipe is 53 in. in diameter, the next 20 ft., carrying 90 per cent. of the air, should be 51 in. in diameter. Treating each successive section in the same manner, its diameter and weight may be determined and the total weight of the piping found to be 3922 lb. Then, the yearly total allowance for interest and depreciation on the piping system will be

$$C_{wo} = 3922 \times 0.025 = \$98.05$$

The loss in pressure due to friction will be the same as in the first case considered, or 0.257 in., but the loss due to the velocity will be only about 40 per cent. of the loss, as calculated in the first example, or 0.095 in. The total pressure loss will then be

$$0.257 + 0.095 = 0.352\ \textit{in.}$$

and the annual power cost at $20 per hp.-yr. will be

$$C_{po} = 30{,}000 \times 0.000324 \times 0.352 \times 20 = \$68.50$$

The velocity for the most economical operation will be

$$V_m = 0.7 \times 1950 \left(\frac{98.05}{68.50}\right)^{\frac{1}{4}} = 1540 \textit{ ft. per min.}$$

Another example of the application of the relationship herein presented would be to assume a case of an entire installation of fan, heater and piping. A fair annual allowance on the entire apparatus for interest and depreciation would be 20 per cent. on the first cost of the installation. Assuming that, in order to obtain the most economical operation, the yearly cost of power should be 30 per cent. of the above 20 per cent. allowance, makes the yearly power cost equal to 6 per cent. on the initial cost of the installation. Allowing $20 per hp.-yr. as the cost of power,† with a 25 per cent. allowance in the cost of the heater for interest and depreciation, the velocity through the heater should be approximately 590 ft. per min. in order to operate at the point of greatest economy.

For the same assumed conditions as to air handled, the resistance through the heater will now be 0.145 in., and at the most economical velocity through the piping system, the pressure loss will be 0.31 in., making a total pressure at the fan of

$$0.145 + 0.31 = 0.455 \textit{ in., or } 0.254 \textit{ oz.}$$

The total annual cost of power will then be C_p = $98 instead of $212.50 under the first assumed conditions when using one straight run of pipe 200 ft. long. This would show a saving of $114.50. The total cost of the apparatus, as at first assumed, would be approximately $1331, so that the power cost of $212.50 would amount to 16 per cent. on the initial cost of the apparatus.

The total cost of the apparatus, as revised to meet the more economical conditions, would be $1660, or an increase of $329. The yearly cost of power would be 5.9 per cent. on the initial cost and there would be an allowance of 19.7 per cent. for interest and depreciation on the entire installation. The saving of $114.50 would make a return of 35 per cent. on the additional investment.

CORRESPONDENCE

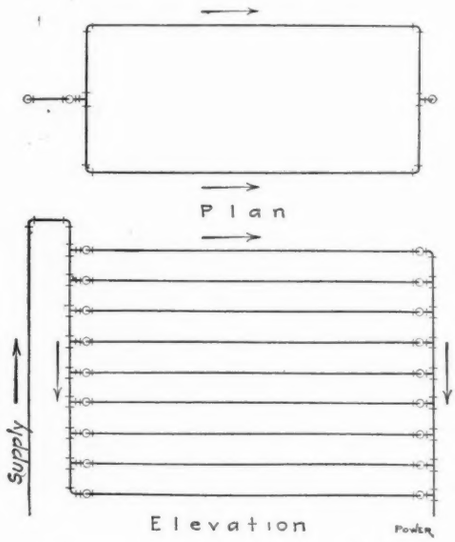
Circulation through Shunted Hot Water Radiators

On page 58 of the Feb. 4 issue of POWER, Mr. Brewster's explanation to the effect that the circulation is due to the cooling of the water in the radiator producing a difference of static head between the flow and return connections is entirely wrong. This would only be possible if the flow-supply connection was covered up to the point where it enters the radiator, which is not the case in practice. As a matter of fact, the flow connection would cool along with the radiator and there would be no flow at all if the radiator were connected to a tank as described. When the system is first started up and the radiator and both connections are full of water at room temperature, what is going to cause the water to start circulating if the circulation is due to a "thermo-siphon" action?

Mr. Brewster also states that the pipes should be so proportioned that the flow does not exceed 10 ft. per sec. The following is a list of pipe sizes with the friction head per 100 ft. of pipe, with the water flowing at the rate of 10 ft. per sec. This shows the large friction head that would result if this practice were followed. As a matter of fact, the best practice does not exceed 4 ft. per sec. in the smaller sizes of pipe:

Size of Pipe, in.	Head in Ft.
1	54
1¼	38
1½	32
2	23
2½	18.5
3	14.3
3½	12
4	10.5
5	8
6	6.4
8	4.7

Another point which is overlooked by a good many engineers in laying out hot-water work, and which is the case in the article under discussion, is that in order to equalize the system, the water in the supply and return pipes should flow in the same direction. If laid out this way, the difference in head is practically the same between the supply and return end of each loop. If laid out with the flow in the opposite direction, the difference in pressure between the supply end of the first loop and the return end may be several pounds, while that between the supply and return ends of the last loop is only a few ounces. The illustration shows what is probably the best method of laying out the loop system, such as described by Mr. Brewster.



PROPER ARRANGEMENT OF PIPING

The reason for the circulation of the water throughout the risers and loops in this system is the same as fo any other hot-water system, the difference in density between the water in the supply and the return risers. The circulation in the radiators is due to the difference in pressure or head between the supply and return end of the radiator, due to the friction loss in the main pipe or loop between these points. This applies equally well to either a gravity or forced system.

W. L. DURAND.

Brooklyn, N. Y.

†See curve B, Fig. 2, page 381, Mar. 18 issue of "Power."

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The Hamburg-American Liner "Europa," sister ship of the "Imperator," is to be launched on Apr. 3, according to recent press dispatches. The "Europa" is 950 ft. long and has a beam of 100 ft.

EDITORIALS

Sleeping While on Duty

In this issue, page 461, a contributor tells of a night engineer who after heavily covering the furnace fire and adjusting the voltage and city water supply to the feed-water heater, would "turn in" for a few hours, leaving the plant in care of a street watchman, who came in hourly, if he was not delayed.

Those who have worked the night watches know how hard it is to resist the desire to sleep which is likely to come over one, especially between two and three-thirty in the morning. Some know, too, that it is an unforgivable offense for any member of the night crew to select a corner and sleep without first informing the man on watch of the particular corner selected. This is necessary to prevent delay in finding assistance should any be needed.

The above is typical of some men and plants. Needless to say it is wrong and the penalty should be immediate dismissal. There may be times when the offense is excusable, but such occasions are rare and, furthermore, the management should be such that it is never excusable.

It is such conditions as herein mentioned that make boiler and license legislation imperative.

❖

Ammonia for Hot Bearings

Undoubtedly the discussion brought out by the letter in the Feb. 18 issue, page 234, on liquid ammonia for cooling hot bearings, will prove interesting.

The rapidity with which liquid ammonia absorbs heat while evaporating commends it for cooling hot bearings. But there is a possibility of cooling bearings and journals too quickly by its use, although as that depends upon the strength of the solution it should be possible to easily control the rapidity of the cooling process.

It should be remembered that rapid cooling of a bearing is only necessary until the temperature has been reduced below that at which the metal tends to become plastic. The aim is to prevent it getting so hot that the babbitt will melt or the brasses become badly "burred" or roughened and so accelerate the temperature rise due to increased friction.

Ammonia is dangerous and unpleasant to handle and for that reason it may never, among many engineers, become a popular means for cooling hot bearings. However, when an engine simply must be kept running, even though it develops a hot bearing, which all ordinary remedies fail to cool, an engineer would gladly use liquid ammonia, if he knew before hand that more serious trouble than a hot bearing would not develop as a result of using the ammonia.

The practice of using a paste composed of white lead, engine oil, graphite and ammonia to "wear down" a chronically hot bearing or pin is not uncommon when the bearing and rotating surfaces become hot because of abrasions on their surfaces. This paste is so effective, even when the ammonia content is small, that it ought to cool extremely hot bearings if the proportion of ammonia contained is increased.

Before applying liquid ammonia to a hot bearing one would like to know how strong a solution is best to use; if the bearings were of brass or bronze would the ammonia seriously corrode them? If so, what could be mixed with the liquid to offset corrosion but not impair the cooling properties of the ammonia?

We would be pleased to publish letters stating the results of actual experience with liquid ammonia as a remedy for hot bearings.

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License Examinations

Quite frequently we receive letters inclosing lists of examination questions that have been asked applicants for engineers' licenses. In some instances the correspondents indicate a belief that if they could answer these particular questions they would have good chances of passing.

The purpose of holding examinations is to obtain evidence of the applicants' fitness for assuming the responsibilities that go with a license. While sample questions may be of value in suggesting the breadth of information and degree of intelligence covered by an examination, an applicant is likely to be disappointed if he has persuaded himself that preparation for an examination consists in simply uncovering correct answers to questions asked at previous examinations. The field of information in which a licensed engineer should be well qualified is so broad that only lame or lazy examiners would adhere to a stereotyped set of questions, hence memorizing answers to old ones is likely to be of little more benefit in preparing for an examination than committing to memory a table of logarithms.

An applicant should remember that the purpose of the examination is not to discover how well he can answer a list of questions pertaining to his work, but, rather, to find out how well he is versed in a knowledge of his work; not how well he can take care of a certain part of a plant, but how well he can care for the whole of it.

To many practical men, some questions asked by the examiners seem foolish or irrelevant. For example, "How many cubic feet of steam at atmospheric pressure weigh one pound?" Although the work of the operating engineer does not compel him to use steam tables very often, his answer to this question immediately indicates whether or not he knows of the existence of such tables and how to use them. The same applies to questions relating to finding the strength of boiler joints or the stress on flat surfaces supported by boiler stays. If the applicant can figure such problems, the examiner knows that he is quite familiar with boiler design and a desirable applicant.

Above all, the applicant should remember that to get his license he must show that he has a good knowledge of boilers and how to safely operate and maintain them. The rest of the plant equipment is secondary as far as his license is concerned.

Is It Worth While?

Routine work like the daily operation of a long-established power plant tends to raise a question in the minds of thoughtful and ambitious engineers as to the ultimate reward of faithful service. Is the game worth the candle when all its trying and wearisome tasks are considered and balanced against the pecuniary returns, present and probable, and the inner compensation which goes with duties well done?

In every field of human endeavor these times of questioning and of temporary discouragement now and then assail the most enthusiastic and energetic workers. Those on the inside of every occupation know that enduring success can only be secured by willingness to endure drudgery and a lasting determination to make each day's record measure a little closer to the highest standards. Thousands of men and women are forced to toil at tasks not of their own choosing to keep body and soul together; but the engineer who has the slightest interest in his work and the smallest natural aptitude for machinery has no excuse for numbering himself among these unfortunates.

The work of the engineer, in this wonderful century of power, covers so vast a field and touches the borders of so many unsolved problems that, in a sense, the money side of these useful and responsible occupations stands apart from their real interest. Happily the possibilities of large financial rewards encompass really distinctive and pioneer engineering work on all sides, but, to the man working for small wages, who has the imagination to see it, the study of his own special installation and the interest which it yields to intelligent analysis often equal the intellectual satisfactions of handling larger and more showy groups of equipment. It is not always the biggest plant which offers the greatest interest to the keen student of power economics.

The engineer working for small pay may be inclined to scoff at the idea that the mental satisfaction of dealing with old mother nature at first hand through steam, steel and electricity enhances the attractiveness of his occupation. With some disgust he may make the observation that mere ideas pay no rents or grocery bills. That is just where he is wrong. The world is looking for purposeful thoughts, pays cash for them when they are put in the right place, and only specifies that, if they are to be taken at par value, ideas shall be capable of practical application along the line of showing how something may be done better than before.

Yes, it is worth while to be an engineer, to put one's best efforts into a plant month after month, if one learns how to look beyond the immediate task and catch its larger significance in relation to the great world of applied science. Some men never catch this spirit of inquiry, never become possessed of an insatiable curiosity to find out the uttermost possibilities of their equipment and to run down and pin to earth the unusual and unexplained events which crowd thick and fast upon every keen seeker after the truth. The work as a whole must be congenial and interesting in its possibilities; if it is not so the engineer had better be selling life insurance or snipping off dress goods behind a counter, or whatever else appeals to him in a permanent way. But once let this fascination of machinery and power seize him, once let the possibilities of self-improvement through spare-hour study and the future possibilities of investigation, and invention take hold of him—then there can be but one answer to the question at the head of this page.

These are great days, and great opportunities for raising the efficiency of some detail of the world's work are at hand. For the engineer who can catch the broader outlook and make it grow by wider association with his fellows, by determined and persistent reading, by seeking after the hitherto neglected or overlooked aspects of his daily duties, by taking upon himself larger conceptions of practical life at the throttle and indicator, his occupation must inevitably be worth while, for its future only awaits unlocking by the master key of intelligent, imaginative and untiring personal industry.

A History of Technics

Elsewhere in this issue will be found an abstract of a communication to the American engineering societies from Dr. C. Matschoss, of the great German engineering society. In it he urges the coöperation of engineers in this and other countries in the preservation of records that may go to make up a fitting history of engineering achievement, or, as he terms it, a history of technics. As he points out, it is evidently because engineers have been so busy doing things that they have had no time to write about them from the historical point of view. Now the danger is that, as these men pass on, there will be lost with them the knowledge of the development of their inventions and many facts that would be of great importance to a comprehensive history of things mechanical or technical.

True, at the time, there are reports upon their work given in the form of papers to the technical societies concerned with their fields of activity, and the technical periodicals give accounts of their work, but these soon become buried in libraries of reference if not altogether lost. What is needed is the work of collating and filling in the history scattered through these thousands of volumes and private manuscripts so that there may be handed down to future generations a complete, coördinated and connected history that may truthfully show the development of the mechanical arts and industries and the evolution of their products.

A condition that is well deserving of the thought of engineers is that history has so long been considered a story dealing with wars and politics. As Dr. Matschoss points out, there are hundreds of such histories while there are none dealing with engineering advance. This is naturally the outcome of the periods when conquest was the paramount concern of nations and civilization was a march of the sword. Present-day development has undergone a radical change even from that intermediate stage when statesmanship took precedence over generalship, for now both warriors and statesmen have largely yielded place to scientists, inventors and that great class popularly known as captains of industry. They are present-day makers of history and their works the salient factor in the extension of civilization and commerce.

Now it remains for a new set of historians to come forward and write up that chapter in the world's history dealing with the events of this age of science and invention; in short, as our German brother has called it, the History of Technics. Where shall we find a Herodotus, a Livy, a Macaulay or a Bancroft who shall be to this age what the older historians were to their times?

READERS WITH SOMETHING TO SAY

Short Splice for Rope Drive

The rope drive under my charge is small, but it gets hard service. A four-strand cored manila rope, 1¼ in. in diameter, is used. The pulleys are 93 ft., center to center, and the drive runs 1600 ft. per min. on an angle of 45 deg. A sliding idler pulley with weights takes up the slack. This drive is out of doors and runs during all kinds of weather the year round. Although a good rope dressing is used, I find that stone dust, which is continually flying about, works its way into the strands and gradually cuts them. In time the core gets ground nearly to a powder, caused, I believe, by bending around the pulleys and by the strain due to the weights on the idler.

A new rope lasts about 2½ years, after that the strands begin to break until the rope is very hard to repair. A splice generally runs from four to six months before giving out, probably because the splice is a little looser than the rest of the rope, which allows stone dust to work in and, the strand, getting the most strain, gives first. A temporary fastening of the loose strand will sometimes allow the plant to continue running until a rainy day or Sunday gives the necessary time to make a permanent splice.

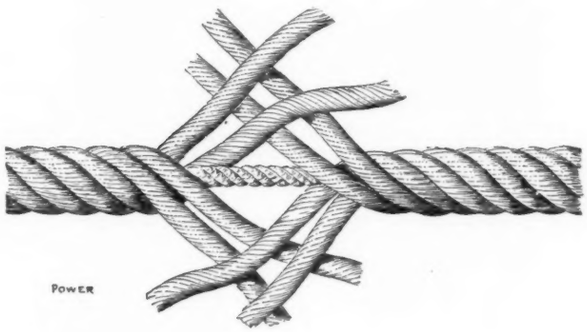


FIG. 1. CORE SPLICE

I use the short splice, taking 2½ ft. on each end which is unstranded after cutting off the core within 6 in. of the solid rope. One strand is unwound, the core pulled out and a short common splice is made, Fig. 1, more to hold the rope together and get the proper twist to the strands than because it is needed. Replacing the core and loosened covering, a strand from each end coming together with the proper twist, is taken and each is split in halves. Taking one-half of a strand it is wound into the other strand, leaving out a thread at each turn, 1½ in. from the other, until both are used up, cutting off all the loose ends. The operation is repeated with each strand until both ends are woven into place. With a little care one can make the pull on each strand nearly equal and in two or three days the splice will be no bigger than the balance of the rope. Fig. 2 shows the method of making a short splice.

One reason I favor a short splice is that it takes very little rope, and two or three splices can be made before it is necessary to put in a new piece, which means a lot when one takes into consideration the time required to make a double splice which is required for the long splice.

I have used the long splice, but it did not last as long as the short one and takes as much time to put in. It takes on the average about two and one-half hours to make a splice with one helper. A prepared rope dressing is used which is applied as hot as can be borne on the hands. It is rubbed in carefully and, although it is a nasty job, there is the satisfaction of knowing that the rope is thoroughly covered. No trouble is had from slipping even when the rope is covered with snow and ice.

C. L. MALLERY.

Myerstown, Penn.

Taking Chances

How some engineers can take chances, and still avoid disastrous accidents, is to be marveled at, considering the way some plants are handled.

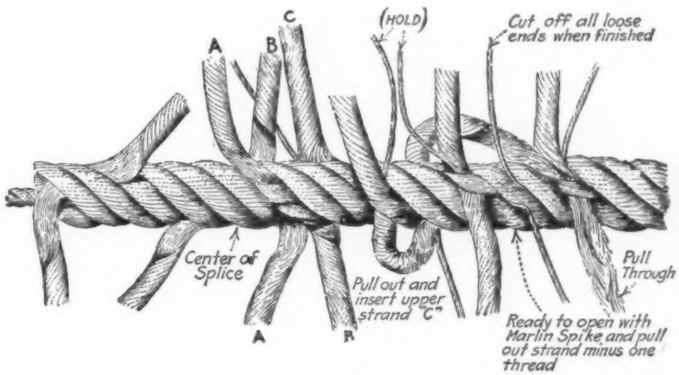


FIG. 2. SHORT SPLICE

At a plant with which I was well acquainted, the night engineer came on duty at 7 p.m. and continued on duty until 7 a.m., seven nights a week, the year round. The plant had two high-speed engines of the single valve type, with automatic flywheel governors, two return-tubular boilers, 72 in. in diameter and 16 ft. long, two duplex pumps, a system of vacuum heating, with two single-cylinder, double-acting air pumps, an electric elevator, a hydraulic elevator and an open heater. The night engineer did his own firing. The elevators were shut down for the night at 8 p.m.

Going over everything, after the heaviest of the load went off, which was about 9 p.m., the engineer adjusted the valve on the city water line to the heater. He then covered the fire heavily, shut the cylinder oil off the engine that was running, pulled out his cot-bed, and "turned in." (He shut the cylinder oil off the engine to change the position of the governor, as the springs squeaked at a certain position and kept him awake.)

A watchman, who was employed by the city, but who had nothing to do with this plant or any of the buildings connected with it, frequently came in during the evening to talk for a while. This watchman made his rounds at 11 p.m., 1 a.m. and 3 a.m. He came in and covered the fire, looked the engine and switchboard over, adjusted the voltage, if needed, saw that the vacuum and feed pumps were running all right, took a final look at the gage-glass and went out again. This state of affairs continued for a year, that I am aware of, without serious mishap.

In another plant, having a 60-in. return-tubular boiler 15 ft. long and carrying a pressure of 90 lb. gage, the engineer went home every Saturday night, leaving a banked fire. He did not come in Sunday, but started a new fire every Monday morning at six o'clock, with the water down in the water column so low he could not find it. This boiler was fed with an old injector which would not work with less than 50 or 55 lb. of steam, and not having any city water connection, he had to get 50 or 60 lb. of steam pressure on the boiler before he could put any water into it. By forcing the fire and getting the water in sight in the gage-glass, he would have about 70 lb. of steam at seven o'clock. He would then blow the whistle and start the engine. The boiler was never blown down Monday mornings, as there was never water enough in it to allow of it.

When asked why he did not come in Sunday to pull over the banked fire and put some water in the boiler, he said: "Oh, I live about 25 miles away, and I go home every Saturday and don't get back until about one o'clock Monday morning, so I take a chance on it."

JOHN C. CHESTER.

Concord Junction, Mass.

Recording Transmission Dynamometer

In the familiar type of rotary transmission dynamometer, in which a spring in torsion is used for transmitting and weighing force transmitted from a driven to a driving element, the problem has been to accurately measure the torsion angle of the spring while the instrument is in rotation. The writer has devised a simple mechanism which measures and records this angle, and with the original model illustrated in Fig. 1 some very accurate power measurements have been made.

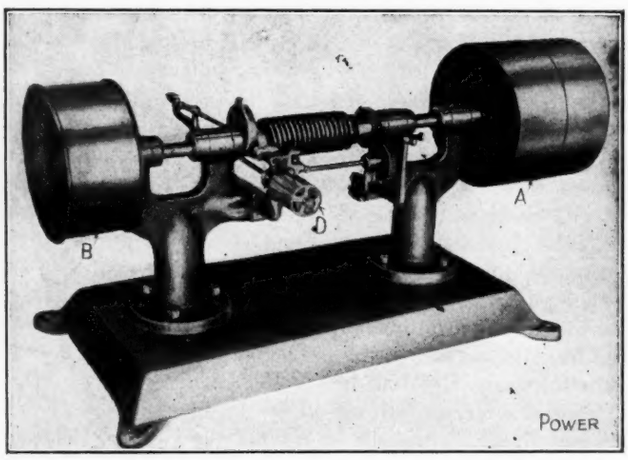


FIG. 1. RECORDING DYNAMOMETER

As may be noted in Fig. 1 the dynamometer is of that class in which the power to be measured is transmitted by a belt to a receiving pulley, *A*, on the dynamometer, and is measured as it is transmitted through the apparatus to the delivery pulley *B*.

Through gears the driving shaft rotates a drum *D* carrying an indicator card, the drum having the same angular velocity as the shaft. An eccentric mounted on the driven shaft moves a pencil back and forth across the face of the drum, giving it a simple harmonic motion, and as the shaft revolves, a sine curve is traced on the card. By allowing the driven shaft to run free this curve will be traced and retraced, giving a reference line or no-load curve. Then applying the load to be measured to the delivery pulley, a second curve, out of phase with the first, will be traced as in Fig. 2.

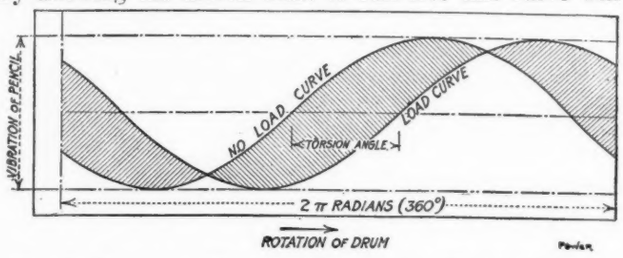


FIG. 2. DIAGRAM MADE BY RECORDING DYNAMOMETER

The horizontal distance or displacement between these curves will be the torsion angle of the spring in radian measure.

Knowing the torsion constant of the spring and the revolutions per minute of the dynamometer shafts, the number of horsepower transmitted can be determined from the formula:

$$hp. = constant \times displacement\ in\ inches \times r.p.m.$$

If the load varies during one revolution of the shaft, the mean displacement must be found and applied to this formula; or better, since the area included between the curves (shown by the cross-hatching) is proportional to the work transmitted through one revolution, the number of horsepower can be computed from the formula

$$hp. = a\ constant \times area \times r.p.m.$$

The metallic pencil used to make the record is held clear of the card by a cord passing to a spring push-button by means of which it is lowered at any instant a reading is desired.

D. D. HUYETT.

Waynesboro, Penn.

Pump Suction Pipe Too Small

An experience I had some time ago goes to show how trouble can be caused by not doing work about the engine room properly.

I was called upon to locate trouble in a pump, and in looking over the outfit found that the suction pipe had been reduced from 2½ to 1½ in. The engineer informed me that the pump had been overhauled, and it certainly was in good condition. But when the trouble was pointed out to the engineer he would not believe that it had anything to do with the action of the pump.

A 2½-in. pipe, as called for by the pump connection, was run to the well from which the feed water was taken. When the pump was started it gave no trouble, and since has been giving the best of satisfaction.

J. R. HARDISTY.

Grand View, Can.

QUESTIONS BEFORE THE HOUSE

Traps and Piping of Return Systems

With interest I read Mr. Little's letter in the Jan. 14 issue, under the heading, "Points about Steam Traps." Mr. Little has laid his finger upon one of the most important features of a proper installation for piping condensation returns. It has often struck me as extraordinary that it has been the common practice of plumbers, for years, to use the 45-deg. ell and Y-connection in the drainage of buildings, whereas the steam engineer still persists in using the old square tee. Manifestly, to properly drain any system having much condensation, friction should be eliminated as much as possible, by keeping the returns flowing in the general direction of the point of discharge.

Proper arrangement of the piping will not only reduce friction but also, very materially, interference of pressures. This method of connecting returns from steam traps has been long advocated by experts on condensation, and many large trap installations have been made in this manner. If this method of piping is applied to a series of kettles and the discharges connected into a common heater, with check valves close to the "Y's," one steam trap on the series is often just as efficient as a trap on each unit of the series. Differences in outlet pressure, as high as 5 lb., are commonly found where kettles and other apparatus are working under the same initial pressure, but at different stages of the same process. Such differences will be equalized in the header and the condensation discharged, if the method of piping advocated by Mr. Little is used.

Where a return trap is used, and the system is kept closed to the atmosphere, it is practically impossible to obtain the best results without this style of piping and the use of intermittent flow traps, with positive discharge, on the apparatus to be drained. Unfortunately, what Mr. Little says is very true, namely, that it is only now beginning to be realized what a great saving can be effected with proper care in the design of the return system, by increase in the efficiency of the drainage system.

I do not agree with him in his praise of the expansion trap, which, in my experience, is difficult to operate under modern conditions of high-pressure steam, and, particularly, where there are large volumes of condensate at a temperature much above 212 deg. F. The best results, I find, are obtained by tilting traps, where the action of the traps is positive and intermittent, irrespective of the temperature of the condensate. It certainly does not pay to return to a hotwell or open heater condensate which leaves the apparatus at any temperature above 230 deg. F.

The main points to be considered are first, the rapidity of drainage, upon which depends the efficiency of the apparatus, and, secondly, the temperature at which this water is available for return to the boilers.

H. L. PEILER.

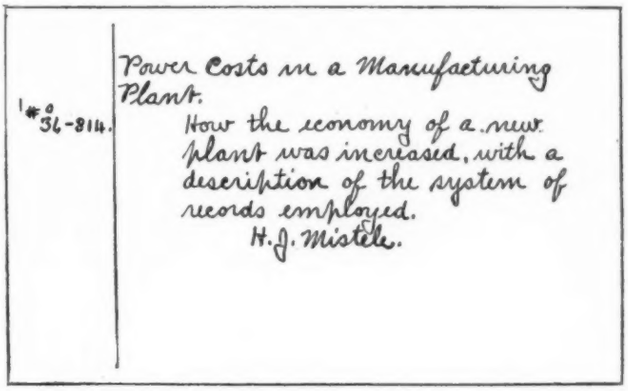
Montreal, Can.

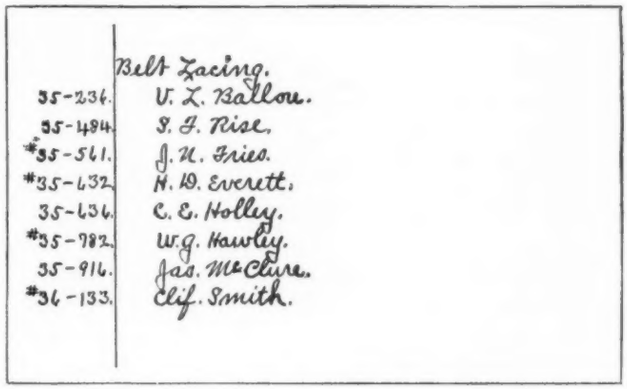
[Mr. Little did not praise the expansion trap; but remarked that, while it was a simple and sensitive form of trap to operate (to quote Mr. Little) "and gives good results, the conditions must be as near ideal as possible."—EDITOR.]

Card Index System

After reading the article by Mr. Thompson, in the Dec. 3 issue, under the above heading, it occurred to me that an index system I developed for myself may prove of interest.

While Mr. Thompson's system answers the purpose very nicely, the one I am keeping is somewhat more elaborate. I am using the 3x5-in. horizontal-bar card which costs $1.20 per thousand. I have the indexed card such as Mr. Thompson uses, and am cross-indexing this system; that is, if a given article such as "Power Costs in a Manufacturing Plant" be taken, I make a complete copy on five cards, placing one back of each of the indexed


Power Costs in a Manufacturing Plant.
#36-814.
How the economy of a new plant was increased, with a description of the system of records employed.
H. J. Mistele.



cards marked "Stations, Description of," "Condensers," "Records and Filing Systems," "Fuels" and "Lubricators and Lubricants." I only use one article or series of articles to a given card.

The numbers at the left of the cards give the volume and page number respectively. When the article is illustrated, I inserted a small figure (#) as shown; or when

an article is accompanied by a table which may prove useful, I insert a different figure (°), as shown on the cards.

JOHN C. SWAIN.

Indianapolis, Ind.

Ammonia for Hot Bearings

A letter in the Feb. 18 issue, from C. E. Anderson, suggested using ammonia on hot bearings. I have never actually tried it, but I know that the advantage as far as cooling is concerned would be offset by a disadvantage not considered by the writer of the letter—the destructive effect of ammonia upon copper and bronze. Therefore, if the bearing to be cooled be bronze lined it would be folly to try any such heroic remedy on it. With a babbitt-lined bearing, the same would be true, for babbitt metal contains more or less copper and so is affected by ammonia. Therefore, since ammonia applied to the bearings for cooling purposes destroys them at the same time, it is plainly evident that ammonia will not do as a cooling agent.

FREDERICK HANSA.

New York City.

[Babbitt metal ordinarily does not contain more than 1.5 per cent. copper, the composition in per cent. being: tin, 45.5; antimony, 13; lead, 40; copper, 1.5.—EDITOR.]

There is no question of the ability of ammonia to cool hot bearings, journals, etc., but I think before placing it on the outside of a hot bearing, it would be safer to cool the shaft or pin adjacent to the hot bearing. By cooling the outside of the bearing first it would have a tendency to clamp or bind on the shaft and thereby aggravate the trouble instead of alleviating it. A much better way would be to first flood the bearing with hot water, then oil the bearing and immediately after use the ammonia and water. This treatment would lessen the danger of any breaks.

GEORGE H. HANDLEY.

Newburgh, N. Y.

No doubt C. E. Anderson's method of cooling hot bearings with ammonia and water is good, but in applying it to a crankpin which has become hot enough to heat the disk, care should be taken that the pin is not cooled too quickly unless the disk is cooled at the same time, or a loose pin is likely to result. It must also be borne in mind that ammonia, even a very small per cent. of it, in the water will attack brass, and hence a bearing containing brasses might be more or less injured, though there would not be much danger of this in the few times the average bearing would receive such treatment.

G. E. MADELEY.

Brownsville, Tex.

Central Station vs. Isolated Plant

The arguments advanced by W. B. Lewis, in the Jan. 7 issue, page 33, interested me because several statements I considered at variance with modern engineering practice. He states that one 25-kw. and two 100-kw. units were proposed for an office-building plant supplying its own heat and hydraulic elevator service, the pumps being steam driven, and that the average load during the day was 20 kw. with a short peak load of 150 kw.

This is a much larger equipment than required. One 35- and one 150-kw. unit would be sufficient. It is true that the one large unit might get out of repair, and cause a temporary shutdown, but as the peak load only lasts about an hour a day it would seem advisable not to install so much reserve capacity for such short duration of load. It is everyday practice in many plants to run a lighting unit every night of the year. I have in mind generating units that are in continuous operation from Monday morning until Saturday night, 52 weeks in the year. I have charge of a 75-kw. unit that has run every night for the past 3½ years, except in July and August, when we use central-station service.

In Mr. Lewis' case, for that reason, I consider the possibility of a shutdown during the time that the large unit is required very remote. He states that the cost of the proposed equipment would have been $20,000, which is $88.88 per kw. This is far too high by at least $20 per kw. I am connected with a modern uptodate plant of 525 kw. capacity in four units with a 15-panel switchboard, and the latest improvements, and the total cost for engines, generators, foundations for engines and pumps, piping and switchboard, including main lighting and power panels was $30,800, or only $58.66 per kw.

First-class machinery for his plant ought not to cost over $60 to $65 per kw., which with the three units that he would use would cost about $14,200, and by using one 35-kw., and one 150-kw. unit, or a total of 185 kw. instead of 225, the cost ought not to be over $11,700. Assuming the first cost to be $14,200, and allowing 5 per cent. for depreciation, 5 per cent. for interest, and 2 per cent. for repairs, the fixed charges would be $1704 instead of $2400, as he figured, and the fixed charges per kilowatt-hour 3.4c. compared with his 4.8c. If the 185 kw. plant was installed the charges on the investment of $11,700 would be $1404, or 2.8c. per kw.-hr. Further on he says, "Statements that exhaust steam is a great factor in lessening the cost of power can be easily overestimated," but in reality its value is usually underestimated by the central-station representatives to make their own proposition look more favorable.

If enough exhaust is not available for heating during the day, it would at least greatly decrease the amount of live steam required, and I fail to conceive of a condition during the heating season in which no exhaust steam would be required for heating during the peak-load period, which he says occurs about 5 p.m.

The building containing our plant is of granite, four stories high, and covers 200,000 sq.ft. of area. With the outside temperature at any point below 46 deg., all of the exhaust steam is used from the average day load of about 130 kw., and a daily peak load from 3 to 7:30 p.m. of 165 kw. On special occasions we get a short peak load of about 250 kw., and even then we have one more 150-kw. unit than necessary, which increases the overhead cost of current to above what it should be.

The building mentioned by Mr. Lewis will require no more labor with a plant installed, as the boilers and pumps must be operated for the elevator service, and the only extra expense will be for coal, oil, etc., which the decreased cost of current will soon pay for, and if current was sold to the tenants at the same price that the central station charged, there would be a neat profit on the ledger at the end of the year, in favor of the isolated plant.

There is another phase of the subject that Mr. Lewis did not mention, and that is the unsatisfactory voltage of the central station which, during the light-load period when few lights are used, is above normal, but as soon as the heavy load comes on it fluctuates, and frequently drops to 90 per cent. of the normal. This is not only unsatisfactory to the tenants, but is hard on the lamps and shortens their life. Frequently there are interruptions to the service of from ten minutes to two or three hours that must be and are overlooked by the tenants, but if these interruptions occurred with the isolated-plant service it would bring down the wrath of everybody in the building on the head of the engineer, and would be advertised from Maine to Texas.

The proposition of a summer central-station service, and a winter isolated-plant service is unsatisfactory. Our institution does it every year, getting current from a large central station during July and August. The home-made current with excessive overhead charges and over $9700 per year for salaries of power-plant employees costs 2.8c. per kw. for a very satisfactory service, and the central-station current costs 2.5c. per kw. for a service that ranges from 90 to 120 volts with a normal voltage of 110, and to which must be added 2c. per kw. for overhead charges on the idle plant and labor that is kept doing a few little repair jobs around the plant that could be as well done while running, making the cost of current 4.5c. instead of 2.8c., and, furthermore, leaving the building with no fire protection except the city pressure. The power-plant men must be kept on the payroll while using central-station current or a new crew hired every year, which causes extra expense and trouble, and the best men are not going to take positions in plants that do not run steady.

There are cases in which central-station service is cheaper in the summer months, but not in the class of buildings Mr. Lewis mentions, in which hydraulic elevators are used, as steam must be carried for the pumps, even though the current is purchased. There might be some advantage if all the pumps and elevators, including the fire pump, were operated by motors.

J. C. Hawkins.

Hyattsville, Md.

Factor of Evaporation

I have just read A. A. Potter's article, in the Mar. 11 issue, on "Factor of Evaporation. His observations as to the heat in a pound of moist steam are correct, but I cannot agree with his conclusions as to the proper method of computing the equivalent evaporation from and at 212 deg.

The items in the Complete Form advised by the committee on boiler testing of the American Society of Mechanical Engineers upon this question read as follows:

57. Total weight of water fed to boiler..... lb.
58. Equivalent water fed to boiler from and at 212 deg.. lb.
59. Water actually evaporated, corrected for quality of steam .....
60. Factor of evaporation..... lb.
61. Equivalent water evaporated into dry steam from and at 212 deg. (Item 59 and Item 60)..... lb.

A footnote referring to item 60 says, "Factor of evaporation $= \frac{H - h}{965.7}$, in which H and h are respectively the total heat in steam of the average observed pressure, and in water of the average observed temperature of the feed."

The above items and footnote are taken from Carpenter's "Experimental Engineering," sixth edition. If the Marks and Davis "Steam Tables" are used, the value for the heat of vaporization of steam at atmospheric pressure will, of course, be taken as 970.4. It will be noted that the factor of evaporation is computed for dry steam, and is multiplied by the weight of feed water actually evaporated, the moisture in the steam being first deducted. Mr. Potter states that this is inaccurate. It is true that the heat of the liquid in the unevaporated water is not included, but this should properly be regarded as lost heat, the same as radiation, heat carried off in the flue gases, and other sources of loss. Moisture in the steam, although carrying heat, is of no use for power purposes, and, in fact, every effort is made to eliminate it by means of separators, bleeders, traps, etc., before the steam is delivered to the engine. The net credit to the boiler, then, should be the amount of water actually evaporated into dry steam.

In his illustrative example, Mr. Potter has multiplied the factor based upon dry steam, 1.13, by the total water fed to the boiler, 5000 lb. This weight should have been first corrected for moisture, making the corrected weight, with his assumption of 3 per cent. moisture, 4850 lb. Also, for accurate work, the factor of evaporation should be carried out to four significant figures. The two methods of computing the equivalent evaporation per hour give the following results:

Water fed per hour, 5000 lb.
Absolute steam pressure, 150 lb. per sq.in.
Per cent. moisture in steam, 3.
Feed-water temperature, 130 deg.

1. Factor of evaporation

$$\frac{0.97 \times 863.2 + 330.2 - 97.89}{970.4} = 1.102$$

Equivalent evaporation $= 1.102 \times 5000 = 5510$ lb.

2. Factor of evaporation

$$\frac{863.2 + 330.2 - 97.89}{970.4} = 1.129$$

Water actually evaporated, corrected for quality = 4850 lb.

Equivalent evaporation $= 1.129 \times 4850 = 5475$ lb.

The latter computation gives a result 35 lb. less than is given by the first method. This 35 lb. from and at 212 deg. represents the heat in the moisture in the steam, which we have rejected.

A. L. Westcott.

Columbia, Mo.

Mr. Flannery's Feed Water Heater Troubles

The Jan. 14 issue contains an interesting letter from W. A. Flannery concerning his troubles with the feed-water temperature. The letter is followed by an editorial note to the effect that the troubles are probably not so much due to improper design as to the manner of installing the heater and to the high solid content of the water.

According to Mr. Flannery's figures there is one boiler at his plant, 48 in. by 15 ft. This would be approximately 50 hp. If it is running at its rated load it is evaporating of water an hour, approximately.

$$50 \times 30 = 1500 \text{ lb.}$$

The heater is 42x100 in. and as the pump supply is 48 in. from the base and the overflow 14 in. above the pump suction, we can figure on a working water depth of say

$$48 + 10 = 58 \text{ in.}$$

The water capacity is, therefore,

$$42 \times 100 \times 58 \text{ in.} = 243{,}600 \text{ cu.in.}$$

or 140 cu.ft. We thus have

$$\frac{140}{25} = 5\tfrac{1}{2} \text{ hr.}$$

supply in the heater, only a little of which is available, however. A heater of this size is usually rated at about 2500 hp.

Regarding the loss due to radiation, the surface of the heater having water on one side and air on the other will be about

$(42 \times 2 \times 58) + (100 \times 2 \times 58) = 16{,}472$ *sq.in.* or roughly, 115 *sq.ft.*

Consider the temperature of the water to be 190 deg. F., as Mr. Flannery says, and the temperature of the boiler room 80 deg. F. By means of Dulong's laws of radiation and convection we find that the heat loss per hour from this surface would be about 19,750 B.t.u., 140 cu.ft. of hot water would be 8400 lb. and

$$\frac{19{,}750}{8400} = 2.35 \text{ B.t.u.}$$

loss which would cause a drop of 13 deg. in the temperature of the feed water.

Large capacity is a desirable feature in a feed-water heater, but when there is over an hour's supply, the temperature begins to be noticeably affected. The fact that the pump suction is 4 ft. above the bottom of the heater and that all this reserve water supply is not available shows poor design of this part of the heater. There is no reason why a properly designed and proportioned open feed-water heater should not give a temperature of 210 deg. F. if there is sufficient exhaust steam to heat the water.

If the water used contains 33 gr. per gal. of solids, it is no wonder the filter becomes choked up in three weeks. This is the penalty for having a bad feed water. I would venture a guess, however, that it is considerably easier to clean the filter than to clean the boiler of an equivalent amount of hard scale.

The regulation of the cold-water supply would be improved if the filter were kept clean. Plenty of hot water for the boiler-feed pump is provided in one well known apparatus on the market by having an automatic bypass which delivers unfiltered water above the filter when the latter becomes fouled so there is always a supply at the pump suction, although only a part of it is filtered. However, I understand this arrangement is patented. Mr. Flannery should not condemn all open heaters on account of his experiences with this particular one, but should see to it that the next one he gets is better adapted to the work to be done. There seems to be little help for him now except to lag the heater to cut down radiation losses, and, perhaps, remove the burlap to allow the water to pass through the filter more rapidly.

EDWARD H. ROBIE.

Philadelphia, Penn.

Mr. Flannery complains of his open heater being too large for the boiler. He is fortunate in having a large heater, the feed water used contains a large amount of solids, and this water, moving slowly through the settling chamber, deposits most of the solids in this chamber and in the filter.

Every open-heater filtering chamber should be cleaned at least every two weeks. I have found wood excelsior, tightly packed into the filtering chamber, with burlap on the top and bottom, an excellent filtering material.

If Mr. Flannery will see that the water distributer at the top of the heater is set level so that the water flows evenly to the perforated trays, also that the rod between the regulating-float chamber and the water-inlet valve is adjusted while the boiler-feed pump is in motion, he will have no fault to find with the temperature of the feed water. It is, of course, understood that the boiler-feed pump will run continuously and that the regulating valve be adjusted to maintain a constant water level—about half a glass—in the heater.

From the present action of the regulating-float chamber it seems to me that the pump suction is too small.

GEORGE MCMILLAN.

Elizabeth, N. J.

Locomotive Boiler Explosions

In the issue of Feb. 4, 1913, I read of another locomotive-boiler explosion at Detroit, Mich., on the Detroit & Toledo Shore Line R.R., on Jan. 5, which was being prepared to haul a train to Toledo. It seems that this boiler exploded under the same condition as the locomotive boiler at San Antonio, Tex., on Mar. 18, 1912.

In the examination report of these boiler explosions, no mention is made of the water being an explosive under certain conditions, the only theory seems to be that of broken stays and some other defective parts, which would naturally be found after an explosion. No mention is made of the water being out of circulation when the locomotive is at rest and no steam is released, and the amount of heat stored equally in all the molecules of the large quantity of water the boilers contain.

Now if as much heat is stored in the water, when a locomotive boiler is in full operation and under the severest condition on the road, why don't the boiler explode? Even when locomotives run into one another and are broken to pieces, no explosion occurs.

To prevent this expansive and explosive force, when a boiler is fired up for service, a part of the steam generated should be released so as to keep the water in the boiler in constant circulation, when there will be no danger to the boiler, which can be proved by demonstration.

Any one that has seen how water acts in a locomotive boiler under different conditions, will disregard the old theory on boiler explosions.

JOHN G. BROMAN.

Chicago, Ill.

The Examiners Examined

Referring to page 292 of POWER, Mar. 4, it would be interesting to have the civil-service examiners examined, to find how they would answer questions 7 and 8, and also to have them define "thermal efficiency" and "thermal per cent."

WILLIAM KENT.

Montclair, N. J.

INQUIRIES OF GENERAL INTEREST

Return Tubular vs. Locomotive Boiler—What advantage has a return-tubular boiler over a locomotive boiler?

W. C.

Return-tubular boilers of good proportions, when properly set and having good draft, are generally more efficient than locomotive boilers because of better furnace combustion and less loss of heat by radiation and by waste heat in the chimney gases.

Fluctuating Water Level—What can be the cause of the water level in the boiler gage-glass fluctuating at every stroke of the engine supplied?

P. A.

When the steam space of a boiler is small in proportion to the size of the cylinder of the engine supplied with steam by the boiler, the water level in the gage-glass may show a variation at every stroke of the engine.

Superheated Steam—What are the advantages of superheated steam?

M. L. E.

Superheated steam, being drier than saturated steam, does not condense or lose heat so readily, and, not being so quickly converted into water, its expansive force is greater than that of steam which is partly water or quickly becomes so.

Safety Valves—What is the largest size safety valve that it is advisable to use?

L. B. G.

It is not well to use safety valves larger than 5 in. because larger ones, especially pop valves, would very soon pound away their seats. Also larger valves are likely to stick and the tendency to stick might not be readily ascertained. It is difficult to get valves larger than 5 in., which will act as sensitively and perfectly as small safety valves.

Water in the Cylinder—Why is the pressure of water in the cylinder of a steam engine dangerous?

J. C. K.

If there is water in a cylinder filling the space on one side of the piston, and the water cannot escape as the piston approaches the end of the stroke, it will result in the same damage in rupturing the cylinder as though the space were filled with an incompressible solid, because water is nearly incompressble.

Boiler Materials—What materials are the proper ones to use for the shells and heads of return-tubular boilers? What is the limit of the tensile strength for each?

O. T.

The shell, drums and butt straps are required to be of open-hearth firebox steel; the heads of openhearth flange, firebox, or extra-soft steel. The tensile strength per square inch for firebox steel is 52,000 to 62,000 lb., and for steel for the heads 45,000 to 55,000 lb.

Loose Piston Ring—If a new snap ring fitted to an engine piston is narrower than the groove so that it knocks at each reversal of the stroke, is there any way to fix it without getting a new ring?

T. A.

This defect can be corrected by taking the ring out and peening it with a hammer on the opposite sides at points that alternate in staggered relation so that the ring is made crooked laterally. This will have the effect of making the ring tight against the sides.

Governing at No Load—With the Rites governor, how can an engine be prevented from overspeeding when without load?

A. B. T.

The steam lap must be great enough to give practically no opening when the governor is in full-speed position, which means zero lead in this position. As the arm swings in, the lead increases, but not enough to give a proper lead in the usual running position unless the governor is set a little ahead. The corresponding disadvantage is an excessive lead at late cutoff, but this is appropriately compensated for best results with the average load.

Countersunk Rivets—Is it allowable to use countersunk rivets on boilers?

G. B. L.

It depends on where they are used. Countersunk rivets are rarely used on a boiler and never should be used to resist any great amount of pressure or strain. Their use is generally confined to places where there is not room for putting heads on the rivets. It is quite common to use countersunk rivets on brass or bronze flanges such as are used for blowoffs, and sometimes on small flanges which are used for steam connections.

Loss of a Governor Ball—What would happen if one of the balls fell off of a Corliss engine governor?

A. B. C.

If the friction of the governor was not too great the engine would speed up. The side twist of the single ball might cause sticking for a short time. When one of the balls is removed practically half of the centrifugal force is taken away, but the arms and connections being left, less than half of the weight is removed, and the weight would overcome the centrifugal force and the collar would drop, allowing the engine to speed up.

Pump to Fill Tank—What size pump would be necessary to fill a cylindrical tank 25 ft. in diameter and 42 ft. high in 10 min.?

J. P. C.

The tank will contain

$$25 \times 25 \times 0.7854 \times 42 = 20{,}616.75 \text{ cu.ft.}$$

This is equal to

$$20{,}616.75 \times 7.48 = 154{,}213.3 \text{ gal.}$$

To fill this tank in 10 min. would require a pump of a capacity of 15,421.33 gal. per min.

Assuming that the piston speed could be 150 ft. per min., a double-acting duplex pump with 36-in. diameter water cylinders by 48-in. stroke would do the work when operating at 18¼ r.p.m. for each stroke of each cylinder would discharge

$$\frac{36 \times 36 \times 0.7854 \times 48}{231} = 211.5 \text{ gal.}$$

and each revolution would mean four strokes or

$$4 \times 211.5 = 846 \text{ gal. per rev.}$$

or

$$846 \times 18\tfrac{1}{4} = 15{,}439.5 \text{ gal. per min.}$$

Locomotive Piston Reciprocation—Does a piston in a locomotive cylinder reciprocate on one end of the cylinder when the locomotive is in motion, or does it reciprocate on both ends?

W. R.

Reciprocation means a motion forward and backward. As motion means the change of position of a body with respect to other bodies, then, considered with respect to any other parts of the engine, such as the cylinder, the piston reciprocates. With respect to the ground, however, over which the locomotive as a whole is moving, the piston does not reciprocate, but has a continually forward motion of variable velocity. This is seen when it is considered that the piston speed relative to the cylinder is never as great as the speed of the whole locomotive, the crankpin not being as far from the center of the driving wheel as the rim. Therefore, even on the return stroke, with the piston at mid-stroke where its velocity is greatest, the piston is not moving backward as fast as the locomotive is traveling forward.

On the forward stroke the piston's velocity with respect to the cylinder is added to that of the locomotive to give the actual speed of the piston relative to the ground. On the return stroke the piston's velocity is subtracted from the locomotive's.

The piston's average speed must be the same as the locomotive's, hence on the forward stroke the piston moves faster than the cylinder and on the backward stroke the cylinder gains again on the piston. This is the basis of the argument that the piston reciprocates in the cylinder on the forward stroke and the cylinder on the piston on the return stroke. Except in respect to each other, neither reciprocates, for, with reference to the ground, both have continually forward motion.

ENGINEERS' STUDY COURSE

MENSURATION—PART VIII
CYLINDRICAL SEGMENTS AND ELLIPSES

The steam space in a cylindrical boiler, as that between the water level *ABCD* and the shell above, Fig. 1, is an example of the form of a *cylindrical segment.*

To find the volume of a cylindrical segment, we must first find the area of the end or base, which is a segment of a circle, and then multiply by the length of the cylindrical segment.

The same reasoning that shows that the volume of a single-base spherical segment is equal to the volume of a spherical sector minus a cone, applies to the area of a segment of a circle. Thus we see that a segment of a circle is equal to the circular sector containing it minus the triangle formed by the chord of the segment as the base and two radii of the circle as sides.

A sector of a circle being only a very great number of thin triangles with their sides adjoining, the area of a sector is equal to the length of the arc multiplied by one-half of the radius, and that of the triangle in question to one-half of the product of the chord of the segment by an altitude equal to the radius minus the height of the segment.

A formula then for the area of a circular segment is

$$A = \frac{a \times r}{2} - \frac{c \times (R - h)}{2}$$

where

a = Length of arc;
R = Radius;
c = Length of chord;
h = Height of segment.

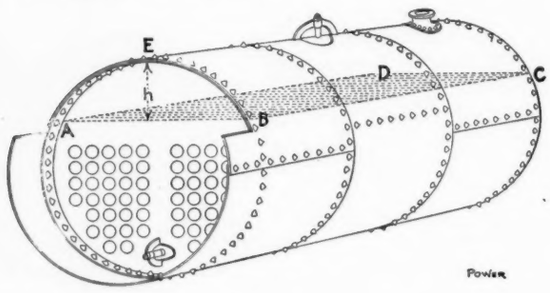


FIG. 1.

Usually the various quantities required in this formula can be measured directly so as to solve the area without any further modifications.

It must be noticed, however, that this formula applies only for segments smaller than semicircles. If larger the minus sign must be changed to plus; then the formula will apply, for the segment of a circle larger than a semicircle is equal to the corresponding sector plus the triangle.

What is known as the Massachusetts formula for the area of a segment of a circle, which is only approximate, however, requires only two dimensions, the diameter of the circle, D, and the height of the segment, h, thus

$$A = \frac{4\,h^2}{3}\sqrt{\frac{D}{h} - 0.608}$$

Various handbooks give tables of segments of a circle of unit diameter, i.e., if the quantities in the table are taken as feet they apply to a circle 1 ft. in diameter or if taken in inches, to a circle 1 in. in diameter. These tables give the areas of segments from a height of 0.001, up to 0.5 (a semicircle), increasing by thousandths. (The height of the segment in these tables is generally called the *versed sine.*)

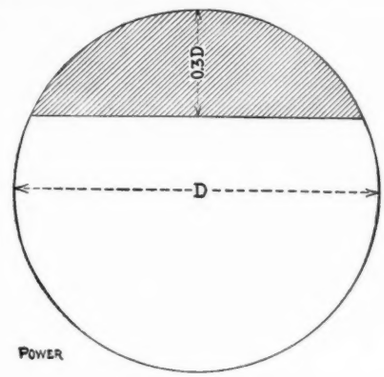


FIG. 2.

Below is given such a table for segments of all heights from 0.01 to 0.5, increasing by hundredths.

TABLE I. AREAS OF CIRCULAR SEGMENTS

Ht.	Area	Ht.	Area	Ht.	Area	Ht.	Area	Ht.	Area
0.01	0.00133	0.11	0.04701	0.21	0.1199	0.31	0.20738	0.41	0.30319
0.02	0.00375	0.12	0.5338	0.22	0.12811	0.32	0.21667	0.42	0.31304
0.03	0.00687	0.13	0.06	0.23	0.13646	0.33	0.22603	0.43	0.32293
0.04	0.01054	0.14	0.6683	0.24	0.14494	0.34	0.23547	0.44	0.33284
0.05	0.01468	0.15	0.7387	0.25	0.15355	0.35	0.24498	0.45	0.34278
0.06	0.01924	0.16	0.08111	0.26	0.16226	0.36	0.25455	0.46	0.35274
0.07	0.02417	0.17	0.08854	0.27	0.17109	0.37	0.26418	0.47	0.36272
0.08	0.02943	0.18	0.09613	0.28	0.18002	0.38	0.27386	0.48	0.3727
0.09	0.03501	0.19	0.10390	0.29	0.18905	0.39	0.28359	0.49	0.3827
0.1	0.04087	0.2	0.11182	0.3	0.19817	0.4	0.29337	0.5	0.3927

If the area of a segment larger than a half circle is wanted, the area of the segment that it would be necessary to add to complete the circle can be found from the table and this subtracted from the area of a whole circle (0.7854) gives the area of the required segment.

For example, in Fig. 2, the area of the shaded segment can be found at once from the table and is 0.19817. If the area of the unshaded segment had been required it would be obtained by subtracting that of the shaded segment from 0.7854 (the area of a circle of radius 1) or

$$0.7854 - 0.19817 = 0.58723$$

If the segment area to be calculated were not that of a unit circle but of one, say, 6 ft. in diameter, it would still be proportional to the value found in the table (areas of circles are proportional to the squares of their diameters), so that it would only be necessary to multiply the value from the table by 6^2.

To find the steam space in the boiler, Fig. 1, by using the table, if the boiler is 60 in. in diameter and the seg-

ment 16 in. high, we would proceed as follows: If the diameter were 1 the segment height would be

$$h : 1 = 16 : 60$$

or

$$h = \tfrac{16}{60} = 0.2666, \textit{ call it } 0.27$$

The area of a segment of such a height, from the table, is 0.17109. Therefore, the area of our actual segment is

$$0.17109 \times 60 \times 60 = 615.924 \textit{ sq.in.}$$

and the volume,

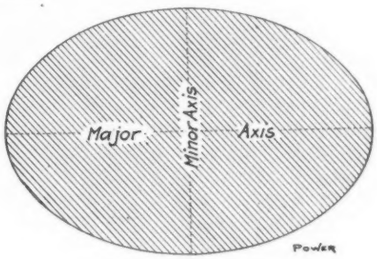


FIG. 3.

$$615.924 \times 18 \times 12 = 133{,}039.584 \textit{ cu.in.}$$

THE ELLIPSE

Fig. 3 shows an *ellipse.* The largest diameter is called the *major axis,* and the smallest diameter the *minor axis.*

The circumference or perimeter of an ellipse can be computed accurately only by the summation of a series and is such a long and involved process that it would be beyond the intended scope of this course to go into it.

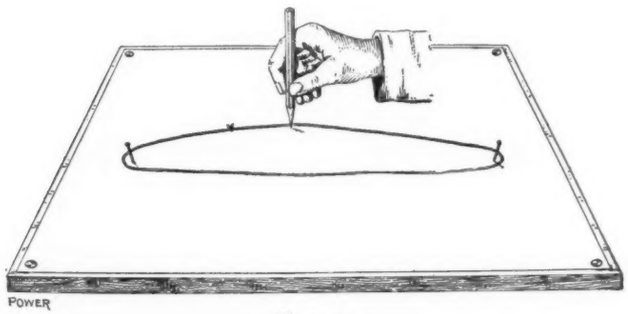


FIG. 4.

Formulas are given in numerous handbooks to figure the circumference with sufficient closeness for practical purposes.

A good approximate formula is

$$C = \pi \sqrt{\frac{D^2 + d^2}{2}}$$

where

$\pi = 3.1416$;
$D =$ Major axis;
$d =$ Minor axis.

Example: What will be the circumference of an elliptical manhole, the major axis being 15 in. and the minor axis 10 in.?

Substituting the values, we have

$$C = 3.1416 \times \sqrt{\frac{15^2 + 10^2}{2}} = 40.047 \textit{ in.} \quad \text{Ans.}$$

The only important things to remember about the ellipse are how to draw one of given dimensions and how to calculate the area.

An easy way to make an ellipse is to stick two pins in a board, as in Fig. 4, drop a piece of string, with its ends joined, over the pins and then, placing a pencil within the loop, draw it taut, as in Fig. 5. Now, if the pencil be held in this relation to the cord and be carried around the pins with its point pressing on the board, it will trace an ellipse. Where the pins are stuck are the focii (plural of focus) of the ellipse and our little experiment shows at once a truth with regard to an ellipse and its focii, namely, that the sum of the two lines drawn from any point on an ellipse to its focii, is always the same. This

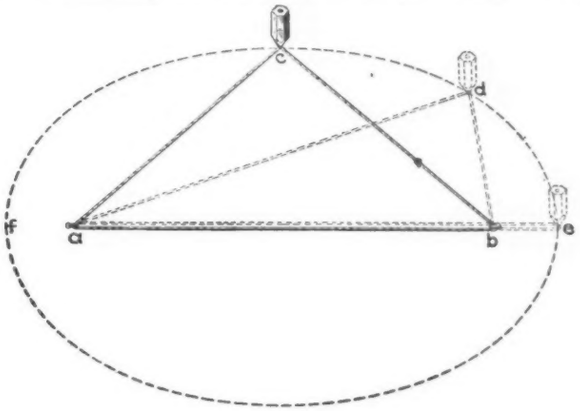


FIG. 5.

must be so because the string is unchanged in length, so that, in Fig. 4.

$$ac + cb = ad + db = ae + eb$$

Another truth then is that this constant sum of the two lines from any point is equal to the length of the major axis, for $eb = fa$. Incidentally we see that the focii are on the major axis.

Now the question is how to draw an ellipse having certain major and minor axes.

First, let us lay out our axes, as in Fig. 6, by drawing two lines at right angles of the required lengths crossing

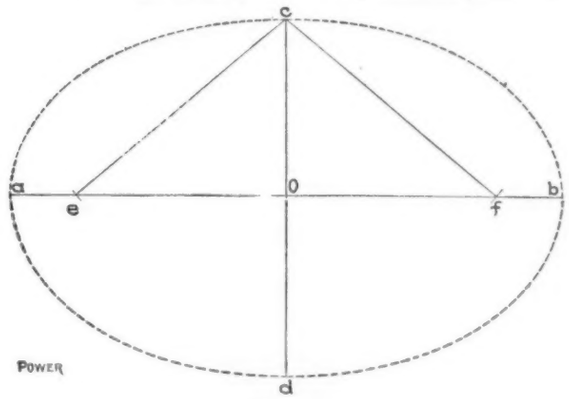


FIG. 6.

each other at their centers, as *ab* and *cd.* We know that the pins must be placed somewhere on the line *ab,* as at *e* and *f,* so that *ec* and *ef* each equal half the major axis. These points can be struck with a compass centered at *c* and set with a radius equal to *ao.* It only remains to stick another pin temporarily in *c* and tie a loop of string around *e, f* and *c.* Then, removing the pin at *c,* we place a pencil point within the loop and trace the ellipse.

Obviously the fatter the ellipse, i.e., the longer the minor axis with respect to the major, the closer the points

e and f will come to each other, until when they come together at o, a circle would be the figure drawn. This circle is known as an *auxiliary circle* of the first ellipse or any ellipse having a major axis equal to its diameter.

In Fig. 7 we have an ellipse and its auxiliary circle. The ellipse was drawn with its minor axis half as long as the major axis. If we draw perpendicular lines to the major axis from the circumference of the circle we will find that the ellipse crosses each one of these lines at its middle and so it would be if the minor axis were $\frac{2}{3}$ or $\frac{1}{3}$ or any other fraction of the major axis—these lines would be crossed by the ellipse at corresponding parts of their length. Incidentally this shows another way to lay out the curve of an ellipse by first drawing a circle and a number of perpendiculars to its diameter and dividing these lines in the ratio of the minor to the major axis.

But this is aside from the point we are leading to—to show the ratio of the area of an ellipse to the area of its auxiliary circle.

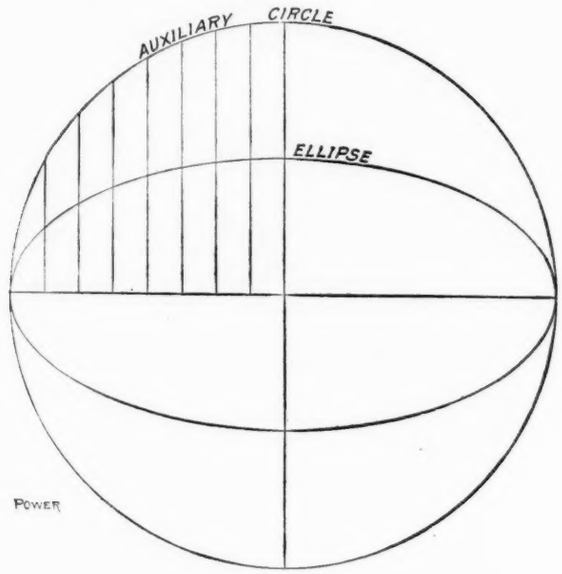


FIG. 7.

If in Fig. 1 the ellipse cuts all the vertical lines at half their height, the area of the quarter ellipse must be half the area of the quarter circle and the whole ellipse half the area of its auxiliary circle. Similarly this holds true for any other ratios of minor axis to major axis so we have the following:

Rule: *The area of an ellipse is to the area of its auxiliary circle as the minor axis is to the major axis.*

Expressed as a formula

$$\frac{A_e}{A_c} = \frac{d}{D}$$

where

A_e = Area of ellipse;
A_c = Area of circle;
d = Minor axis;
D = Major axis = Diameter of auxiliary circle.

We learned before that the area of a circle is $0.7854D^2$.

Then we can substitute this for A_c when our equation becomes

$$\frac{A_e}{0.7854\,D^2} = \frac{d}{D}$$

or

$$A_e = \frac{0.7854\,D^2 \times d}{D}$$

which, canceling the D's, reduces to

$$A_e = 0.7854\,D \times d$$

which is only another way of saying:

Rule: *The area of an ellipse is equal to 0.7854 times the product of the axes of the ellipse.*

Example: What is the area of the ellipse in the previous example?

The major axis is 15 in. and the minor axis 10 in. Applying the rule, we have

$$15 \times 10 \times 0.7854 = 117.81 \text{ } sq.in. \text{ Ans.}$$

ANSWERS TO LAST WEEK'S PROBLEMS

1. $917.66.
2. 1229.88 gal.

OVER THE SPILLWAY

JUST JESTS, JABS, JOSHES AND JUMBLES

Horan—Is it right to write "alright"?

Doran—No, "alright" is not right, but "all right" is right to write, all right. Much of our language is written so rotten that to write right is— Ye get me right, Horan, when I say that many of our wroters have wrotten so rotten—that is, their wrotings being written so unright—

Horan—Say, Doran, do you know what you're sputtering about?

Doran—Well, I did when I started, but—it's a hard language to speak right.

This issue of Power is dated Apr. 1, and is the thirteenth number of the new volume of 1913. A jinx? Not on that crayon portrait of your wife's dear mother! Our only hard luck is that we can't get in one issue all the good stuff that comes in weekly from our correspondents and contributors; also, that we are so dinged modest that we can't print some of the nice things our readers are saying about us.

AND IT ALMOST HAPPENED

Anybody who knows Timothy Healy, national president of the Eccentric Firemen's Association, and a member of nearly everything else, including the Cold-water Recruits and the Boy Scouts—we say, anybody who knows him says he's the divvle at a joke. The other day Tim came into the office with Jack Armour, another boy comedian, and together they put one over on Billy Spills, the engineer of this "Spillway plant." They started in to scare him out of his overalls—and it almost happened.

"Hey, Spills," said Jack, "Tim Healy is out in the library with blood in his eye. He says you wrote a pome in POWER'S 'Over the Spillway' which insinuated that he was trying to break into the Four Hundred by joining the Spugs, or Spuds, or—anyway, come out and square yourself."

Billy made the first hundred yards on one foot, but he nearly dropped when he beheld a fine figure of a man with a red—no, a brown mustache, and red in his eye.

"Young feller," says Tim, "you have traduced the fair name of an honest but hardworking man; you have sown seeds of dissension in his domestic circle, and slaughtered the peace and happiness of a faithful husband and loving father; you have— What do you mean? Answer me quick, before I say something"!— and it almost happened.

"But—but, Mr. Healy—" began the unhappy Billy.

"Butt in with no 'buts'; I've been vilified, slandered"—

"Why, Mr. Healy, it was all a joke!"

"William, I haven't slept a wink since nine o'clock this morning, when I got up, for the mental upset you have given me by your utterly unwarranted and highly inexcusable impertinence. However, for a Swede, you have turned out a fair Irish pome, and—

"Say, Jack, tell him we're only handing him some of his own medicine. It's all right, William. Jack will give you the Cremo I gave him a while ago; I don't smoke that brand myself."

And it almost happened!

Notes on Superheating

By C. R. D. Meier*

In an article by E. D. Dreyfus in the April, 1912, *Electric Journal,* the percentage of decrease in the turbine water rate due to the use of superheated steam is given as follows: 50 deg. superheat, 5 per cent.; 100 deg. superheat, 10 per cent.; 150 deg. superheat, 14 per cent. A paper entitled "Recent Developments in Steam Turbine Practice," by K. Baumann, read before the Manchester section of the Institution of Electrical Engineers, Jan. 16, 1912, gives the following figures: 0 to 100 deg. F. 1 per cent. improvement on the steam consumption for every 10 deg. F.; 100 to 200 deg. superheat 1 per cent. improvement of steam consumption for every 12 deg. F.

This paper also brings out another important point: the efficiency (referred to Rankine) increases considerably with the increase in superheat, the following figures being given: For 100 deg. superheat 2.5 per cent. better efficiency and 6.75 per cent. better efficiency for 200 deg. superheat.

The improvements in steam consumption and turbine efficiency given, due to the use of superheat, are compared to the efficiency and steam consumption with dry saturated steam. Ordinarily, however, steam supposedly dry will contain a small amount of moisture which causes a reduction in efficiency. According to the paper referred to, the efficiency will change 1 per cent. for each 1 per cent. variation in wetness, and, therefore, steam consumption measured as condensed water will be 2 per cent. higher for each 1 per cent. increase in moisture. With reciprocating engines, the presence of moisture in the steam has a still greater detrimental effect as it greatly augments cylinder condensation. For this reason an increase in economy of 12 to 20 per cent. is not uncommon with superheating only from 40 to 100 deg. F. (See "The Steam Engine," by Perry.)

The saving in coal consumption due to the use of superheated steam is not as large as the percentage saving in steam consumption; 100 deg. superheat will reduce the steam consumption of a turbine, say, 10 per cent. This means that instead of 1 lb. of water evaporated under dry and saturated conditions, only $\frac{9}{10}$ lb. of water need be required if the steam be superheated 100 deg. Assuming the feed temperature as 200 deg. the heat absorbed in the boiler for 1 lb. of steam at 150 lb. gage without superheat is 1027 B.t.u. The heat in $\frac{9}{10}$ lb. of steam under the same conditions is 924.3 B.t.u., the difference, 102.7 B.t.u., being saved. To secure this saving, 10 deg. superheat must be added to each pound of steam. According to the latest steam tables, this requires 57 B.t.u. per pound, or 51.3 B.t.u. for $\frac{9}{10}$ lb. This amount of heat, added as superheat, saves twice as much heat, 102.7 B.t.u. in the boiler.

It may, therefore, be said that by using superheated steam there is obtained a fuel economy equal to half the steam saving. This means that heat imparted to a superheater is just twice as valuable and generates just twice as much power as heat imparted to a boiler.

If by using superheated steam, 10 or 15 per cent. of the plant capacity may be generated for just half the cost of the remainder of the output, it is of great importance that the superheat for which the turbine or engines are best adapted be determined by trials and then maintained continuously. The temperature of the steam should not be allowed to fall below the mean, since economy is sacrificed, and, on the other hand, the temperature should not be allowed to exceed the predetermined mean, as troubles with valves, fittings and cast parts are likely to arise.

When the economy of superheating was first realized, superheats as high as 300 deg. were sometimes specified, and superheaters of a good many square feet of surface installed. It was found, however, that steam at these very high temperatures gave a great deal of trouble and caused rupture of cast-iron valves and fittings. By substituting high-grade cast steel, high superheats could be carried, but the added cost for these improved fittings tended to counterbalance the added returns from the use of higher steam temperatures. As a result, it has been found that the highest commercial efficiency, everything considered, is secured by the use of moderate superheat, and it has now become quite general practice to run at from 125 to 150 deg., in turbine plants, and the same or somewhat lower superheat in reciprocating-engine plants, where a low superheat eliminates losses from moisture and reduces cylinder condensation without causing lubricating troubles.

The temperature to which steam will be superheated depends upon the amount of steam flowing through the superheater tubes and its temperature, and secondly upon the amount of gas flowing over the tubes and its temperature. As the quantity of steam is controlled by the boiler load, and the temperature of the saturated steam is practically constant, the degree of superheat added will be determined by the amount and temperature of the hot gases, which will in turn depend on the design and arrangement of boiler and superheater.

There are numerous methods of installing superheaters. First may be considered that in which the superheater is placed in the path of the combustion gases after they have passed over a part of the boiler-tube surface. As the first pass in such boilers absorbs as much as 80 per cent. of the heat in the gases, their temperature when passing over the superheater surface is comparatively low, and hence the superheater surface must be quite large. It is difficult to obtain adjustment or close regulation of superheat with this arrangement. If the volume of combustion gases and their temperature varied always in exact accordance with the amount of steam generated and passing through the superheater, the degree of superheat would be practically constant. But the amount of air used to burn a pound of coal, the temperature of the fire and, therefore, the temperature of the gases after they have passed over part of the boiler surface and come in contact with the superheater surface, vary even with constant boiler load, and since the load varies with boiler pressure and the demand for steam, there are likely to be wide fluctuations in steam temperature when a superheater is installed in this manner.

*Abstracted from the "Journal" of the Engineering Society of Pennsylvania.

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Forty minutes after they had been declared dead from suffocation seven out of twelve firemen recently overcome by smoke and gas during the destruction of the Westinghouse electric plant by fire in Pittsburgh, Penn., were revived by pulmotors at a hospital, and owe their lives to this life-saving invention. The Bureau of Mines has a full equipment of pulmotors for its life-saving crew, and they proved successful in many instances.—"Compressed Air."

Air Supply and the Air Receiver

By Frank Richards

SYNOPSIS—A specific example in air compression, in which a compressor supplies air to an air engine. Also method of maintaining constant receiver pressure.

The following question has been brought to my notice with a request for an answer and, no doubt, the information contained will be of interest to Power readers:

"If an air compressor has a capacity of 90 cu.ft. of free air per min., and delivers it at 90 lb. gage pressure, what volume does the compressed air occupy? My position is this: I have an air engine 3½x3½ in., which is to be run by an air compressor rated to deliver 90 cu.ft. of free air per min. at 90 lb. pressure and 250 r.p.m. The engine will run at 350 r.p.m. intermittently (about 3 min. with 6 min. rest), but the air compressor will be run continuously. What size reservoir would be necessary so that the pressure would not drop too suddenly? I would like to have the calculations on this, if possible. Also, how many cubic feet of free air does it require to give 1 cu.ft. of air at 90 lb. pressure?

A. The question at the end of the above is covered by the one at the beginning, so that one answer will cover that part of it. It is generally better to call the normal atmospheric air pressure 14.5 lb. instead of 14.7 lb. as is customary, because we are generally enough above sea level to make this a better average. Then, if the air is compressed to

$$90 + 14.5 = 104.5 \; lb.$$

the volume of 1 cu.ft., when compressed, will be

$$14.5 \div 104.5 = 0.1387 \; cu.ft.$$

It is not necessary to consider whether the air is compressed isothermally or adiabatically, or what may be its temperature when it leaves the compressor, as it will certainly lose all its heat before being used and then the above figures will apply.

The compressor is said to be rated at 90 cu.ft. of free air per min. This is taken to be the builders' rating based upon the full piston displacement. If all the inefficiencies are to be allowed for, there must be at least 20 per cent. deducted from the theoretical capacity, which would make the actual free air capacity 72 cu.ft. and the delivery capacity would be

$$72 \times 0.1387 = 9.98 \; cu.ft.$$

or, say, 10 cu.ft. per min. at 90 lb. pressure.

The air consumed by the engine at full pressure, 90 lb., and without cutoff will be

$$\frac{3.5^2 \times 0.7854 \times 3.5 \times 2 \times 350}{1728} = 13.64 \; cu.ft.$$

To this 25 per cent., at least, should be added to cover clearance and other losses, making the consumption 17.05 cu.ft. per min., at 90 lb., or

$$17.05 \div 0.1387 = 122.92 \; cu.ft.$$

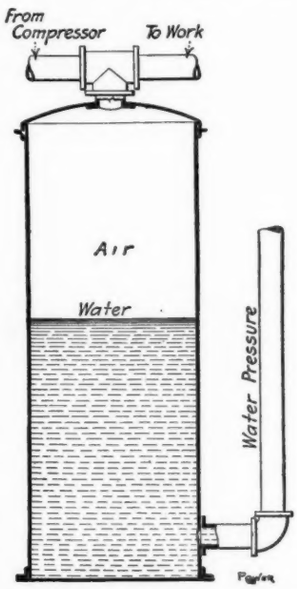
of free air per min.

As the compressor would run all the time, while the engine would run, according to the statement, only one-third of the time, the compressor delivery to supply the engine should be

$$122.92 \div 3 = 40.97 \; cu.ft.$$

of free air per min. instead of 72, and it would be necessary either to slow down the compressor or to blow off some of the air at the safety valve. If the compressor were slowed down from 250 to 150 r.p.m., the free air capacity per minute would be 250:150::72:43. The 43 cu.ft. would be about right for the case as stated.

The receiver capacity would depend upon the amount of pressure drop that would be permitted, and the question does not tell this. Say that the air receiver has a



Uniform Pressure in Air Receiver

capacity of 100 cu.ft., and is filled with air at 90 lb. pressure when the engine starts. The total contents would be

$$100 \div 0.1387 = 720.9 \; cu.ft.$$

of free air, say, 721. In three minutes' run of the compressor, it would put into the receiver

$$43 \times 3 = 129 \; cu.ft.$$

of free air, making the total

$$721 + 129 = 850 \; cu.ft.$$

In the run of three minutes, the engine would take out

$$122.9 \times 3 = 368.7 \; cu.ft.$$

of free air, and

$$850 - 368.7 = 481.3 \; cu.ft.$$

of free air or its equivalent at a higher pressure, remaining in the receiver. Now

$$481.3 \div 100 = 4.81 \; cu.ft.$$

of free air to each cubic foot of receiver content, and 1:4.81::14.5:69.78. The pressure would be 69.78 lb. abs., or 55.28 lb. gage pressure, which would probably be a greater drop than the case would allow.

It is proper to remember that if the engine kept running at a uniform speed, taking out a constant volume of air for each revolution, the pressure drop in the receiver would not be as great as above computed on account of the gradual expansion of the air as the pressure was diminished, but it is evident that it would be enough to defeat the scheme of the installation, and a much larger receiver would be required.

The outcome of the above question is just as unsatisfactory as it always is when dealing with an air receiver

as usually installed. There is no possibility of maintaining a uniform pressure in the receiver if there is any difference in the relative rates of compression and consumption, and as far as power storage service is concerned, the receiver might as well not be.

Constant Receiver Pressure

It happens that in most places there is water under pressure which may be piped to the receiver, city water service, water pumped to a tank on the roof or otherwise provided, and this water pressure may be employed to maintain the air pressure constant in the receiver, whether there is little or much air in it, so that the entire contents may be used at the full pressure, even if the compressor is stopped, or it may be used fast or slow within the limits of air supply with perfect satisfaction.

The arrangement is about as simple as anything in the world can be. As shown in the sketch there is a vertical air receiver, and near the bottom of it an open pipe connects the interior of the receiver with the water supply, so that the water pressure is always the pressure in the receiver. At the top of the receiver is an air pipe connecting the receiver with the compressor and the air being used is taken away from the same pipe which is in connection with the receiver. No valve or contraption of any kind is required, not even a safety valve, for the pressure can never exceed the constant water pressure.

If the compressor is delivering the air just as fast as it is being used none of it will go into the receiver at all but will pass right along to its work. If the compressor is delivering more air than is being used the excess of air will go into the receiver and drive the water down in the receiver and back to its tank or reservoir. If there is more air being used than the compressor is supplying, the water pressure will drive air from the receiver into the pipes to make up the deficiency; and this it will continue to do until the air is all out and the receiver is full of water. There is not even any waste or consumption of water, as none of it is carried off or escapes. It just plays back and forth between the receiver and the water tank or reservoir, as will be evident from the sketch on the opposite page.

If there is a reliable water pressure obtainable and if it is not just the pressure desired, this can be varied by locating the receiver upstairs or downstairs, on the roof or in the cellar. Placing it 11 ft. higher will reduce the pressure 5 lb. and putting it 11 ft. lower will increase the pressure the same amount. With this arrangement so generally available, its employment should add much to the value of the air receiver in compressed-air practice. This device was employed in magnificent proportions by Bruno V. Nordberg for the reconstructed hoisting plant of the Anaconda Copper Co., located at Butte, Mont.

Coal Testing under Difficulties

By B. S. Hanson

SYNOPSIS—It was impracticable to weigh the coal or water, so a comparison of coals was made from the proximate analysis.

A short time ago the writer was called upon to make a comparative test of two different grades of coal. One grade of coal had been used for some years, and the other was a sample of coal submitted by a coal dealer. My employers desired a test accurate enough to show them the value of the sample coal in comparison with the regular coal, but objected to any extra expense in making the test.

At the plant the coal is not weighed, the car weight being accepted without question. The coal cannot be weighed in the boiler room, as it is of necessity dumped from the cart so close to the boiler fronts as to leave no room for scales or any method of measuring. The coal cannot be dumped anywhere else in the boiler room and nowhere out in the yard near enough for convenience. In fact, weighing the coal at the boilers would cause great inconvenience, and the expense of extra help. There is no scale to weigh by the cartload.

Even if the coal had been weighed this data would have been of little use without the corresponding weights of water evaporated. On looking into the matter of getting the water weight or volume, I found practically the same trouble that prevented me from getting the coal weights. The management did not wish to purchase a water meter nor to go to the expense of altering the feed piping and pumps so as to allow the water to be measured by means of barrels.

As an evaporative test was out of the question, I decided to try and make a proximate analysis of the coals. The only apparatus in the engine room was an old pan scale with a set of weights in drams. The dyer had a fine chemist's balance with a complete set of metric weights. It was inconvenient to use the dyer's balance, but by borrowing a spare set of his weights, I was able to test and correct my scale.

The coals were sampled from the cartloads as they came into the boiler room by taking shovelfuls from all parts of the load, mixing and quartering them down on a large iron plate, breaking the lumps while mixing and finally putting a 2-lb. sample in a tin can with a close-fitting cover. As many as eight samples were taken from each coal. Each of the 2-lb. samples was divided into two parts, one of which was weighed on an accurate scale and put over the boilers to dry out; the other half was put through a coffee grinder and ground as fine as possible. This part was also weighed and dried, and then both samples were weighed again and from the loss in weight the percentage of moisture was calculated. The two samples agreed very closely, the fine-ground sample showing slightly less moisture than the other, which was accounted for by the drying out in grinding.

To obtain the volatile contents a sample of the fine-ground coal, of about 3 grams weight, was placed in a retort made of a 1¼x3-in. wrought-iron nipple with a cap screwed on each end and a small hole drilled in one end to permit the escape of the gases. This retort was then heated to a red heat in the flame of a gasoline blowtorch, which was concentrated around the retort by means of a muffle made of asbestos millboard and firebricks. The retort was kept hot for 15 min. after the last trace of gas

had shown at the escape vent. It was then allowed to cool, the coke content was removed and carefully weighed, and the difference between the weights of the coal and coke was taken as the weight of the volatile gases in the sample.

The coke was then crushed and returned to the retort and one of the caps was left off. The retort was heated red again and the coke was gradually burnt to ash, which generally took about two hours. After the ash was cooled off it was weighed, and the difference between the weights of the coke and the ash was taken as the weight of the fixed carbon. The different percentages were then calculated from these weights. In this way I was able to do three analyses a day and complete 24 analyses in a week and a half.

Samples were taken from the coal in the storage piles, which had been in storage six months, from the coal being used as it came in from the cars, and from the two sample cars of the new coal to be tested.

After the analyses were made the results for each coal were averaged and the heat units per pound calculated by means of the following formula:

$$Q = (\textit{Per cent. fixed carbon} \times 146.5) + (\textit{Per cent. volatile} \times \textit{constant } K)$$

For the coal in use K was taken as 180 and for the new coal as 200, K varying according to the percentage of volatile present. Great accuracy is not claimed for this method, but as both coals received the same treatment the results were accurate enough for comparison. At the same time the behavior of the coals in actual use was carefully watched and the amount and character of ash noticed. The behavior of the coals was about what would be expected from the analysis.

The results of the analysis are given in the accompanying table. Coal A was the regular coal taken from the cars, B was the same kind of coal taken from the storage pile and C was the new coal. Coal A and B is a Cumberland deep vein run-of-mine and was then known as boat coal. Coal C was reputed to be from the George's Creek district and was all rail coal. All rail coal is not broken up so much as the boat coal, being handled less. Coal A and B cost $4.45 per gross ton delivered in the boiler room, while coal C cost $4.30 delivered.

PROXIMATE ANALYSIS OF COALS

Coals	A	B	Average of A and B	C
Fixed carbon	66.950	67.757	67.350	67.375
Volatile	22.430	19.648	21.040	15.525
Total combustible	89.380	87.405	88.390	82.900
Moisture	3.050	3.000	3.030	2.650
Ashes	7.570	9.595	8.580	14.450
B.t.u. per lb.	13,845	13,463	13, 654	12,957

Coal A and B is a high-grade Eastern free-burning bituminous coal. It smokes a little and the ashes clinker some but not badly. The coal makes steam well with either natural or forced draft.

Coal C is a fair grade of semibituminous, which smokes very little. The ash content is high, but the coal does not clinker. This coal contains no more fixed carbon than the regular coal and contains 5.49 per cent. less combustible. In comparison with A and B coal C stands as follows in each particular.

	Per Cent.
Fixed carbon	100.04
Combustible	93.79
Volatile	73.79
Moisture	87.46
Ash	168.42
B.t.u.	95.00

In comparing the cost of the coals add to A and B 3¼c. per ton and to C 5½c. per ton for ash removal. Then, for the same heat value as given by a ton of A and B costing $4.4825, there will be needed 1.053 tons of C costing $4.5858, or 10⅓c. more than A and B. With a coal consumption of 4500 tons per year, the increased cost of using coal C would be $464.85.

The practical test showed that the steam-raising capacity would not be as good with coal C as with A and B, as C did not burn freely, needed more draft and made dirtier fires. To burn coal C to advantage and maintain sufficient steam at all times, it would have been necessary to use forced draft at times and to install shaking grates so as to facilitate the removal of dirt from the fire. The increased amount of ashes shown by the analysis of coal C was confirmed by the practical burning test, here the increase in the amount of ashes was so large that the yardman complained about the extra work of removing them.

After receiving the report the manager decided to stick to the old coal. He brought out the specifications given by the dealer who sold coal A and B and it checked with the analysis very closely. The firm guaranteed: fixed carbon, 66 to 68 per cent.; volatile, 22 to 24 per cent.; sulphur, not over 2 per cent.; moisture, 2 to 4 per cent.; ash, 5 to 7 per cent.; B.t.u., not less than 13,500.

System in Charging

By W. H. Booth

SYNOPSIS—A satire on a system of charging for electric current, showing its ridiculous features, by applying the principles in the conduct of a country store.

Many men have worked all their lives in one place. Such a one was Joe Smith, who left our village straight from school and started as junior clerk at the big electric-light works in Blackborough about ten miles distant. He would come home occasionally at week ends and holidays and we had then the opportunity of observing the phases through which he passed by way of high-up and low-down types of collars, turned up trousers, hats of various brims and fancy vests.

His father kept the little general store at the High St. corner and managed to pay his way, but did not amass much of a fortune. Joe sometimes came with him to the inn on Saturday evenings and was wont to air his ideas on what he called matters of commerce. He felt sure that his father was not in the right path financially, but he could not say where the old man was in the wrong, for he had never taken any interest in the store. But he was sure he could make plenty of money by a proper system of charging such as he had learned at the big works at Blackborough. He bragged that the lighting station had made 5000 pounds deficit last year and it must be a paying concern if it could pile up all that.

Joe was not very lucid in his explanations, but we understood him to say that every consumer paid four pence a unit as a beginning if he kept one lamp or any number of lamps alight 24 hr. every day in the year; but if he lighted fewer lamps he paid more for the time the fewer lamps were lit, and if he had an evening party once a quarter, and did not invite the station engineer or the secretary, he paid double for all the current used that quarter. But if he invited the engineer this man would bring a key that would put back the electric meter every

other day not being the fourth day of each alternate week after the party. And he told us of other rules and exceptions and of the army of clerks calculating the fines on the poor widows who did not use enough electricity, and making up the big discounts for the rich men who had parties every night.

The whole business was explained by Joe to be a sort of cross betwixt the system of maximum demand and minimum light and the undulating flat rate constant low-voltage system, and if his father would condescend to learn from him he would put his store books in order and start him on a sound system.

So the old man went off for a holiday and Joe, who was taking a summer vacation, started in to run the store for a week. Joe spent all Sunday looking into the customers' names and books, and was ready for action on Monday, letting down the shutters at 8 a.m.

Old Granny was his first customer; item, a penny bloater—charge, two pence instead of the old price of one penny. Joe explained that Granny only bought two bloaters a week at a high rate of maximum daily demand. Then he sold 60 bloaters for a half penny each to a fellow who had been watching the game and who promptly distributed them at ¾ pence each to a crowd outside whom he called a "syndic." Joe made the curate pay a fine of six pence on his groceries, and explained to him that if he would come in and buy one shilling's worth every day instead of six shillings' worth a week he could have a bonus of two pence a day and so save 18 pence a week by his more level demand.

Mrs. Brown, who had bought a loaf of bread about twice a week, was charged six pence in place of four pence a loaf. The vicar, who was accustomed to buy three or four cases of mineral water at one time, had 35¼ per cent. added to the price, but was told he should call twice a day for a bottle at a time and would then be given a discount of 11⅔ per cent. The vicar had been senior wrangler* at Cambridge and these fine figures were Joe's concession to a brother mathematician.

This sort of thing went on for a week, but whatever else it did, it did not increase output. Joe would come around to the pub after closing time and discuss the question of the strain made by the heavy demand for whole cases of stuff and of the small profit made on one penny bloaters. Sam Jones tried to show him it did not really cost him much to sell bloaters, for people went into the shop and put down a penny and just lifted the fish and said, "Morning" and cleared off. This sort of thing he said went on all day and a crate of the blessed things had gone by night, yet there were never more than two buyers of bloaters in the shop at one time and it was the one-at-a-time buyers who got rid of the crateful and not the three families who bought each six fish a week.

Joe positively refused to sell a hundredweight of potatoes to the manager of the brush factory, who had a big family and bought things in fairly large quantities. "It didn't matter," said Joe. "So much the worse for a man if he bought a ton at a time, for his demand factor was low and his maximum demand was cruelly high." The big families in the town, before the week end, were sending their children every half hour for an egg or half an ounce of salt and a dozen beans and Joe was run off his legs keeping pace with the demand.

During the first two days his receipts had increased 50 per cent., but during the last four days, when people had learned the value of a steady purchase rate, the output of goods went up considerably, the receipts fell to about three-fourths and the profits to zero. Joe had achieved the deficit of which he was so proud in the case of a big station at Blackborough. The old man was very patient about it. He told Joe that he liked the little customers, "They mounts up in the course of a day and I learns things from them, and they keeps me alive."

The chairman of Joe's company lives in our village, or in a big house just outside, and he had seen all that had gone on in the week of Joe's experiment. It is reported that he has decided to change the system of charging at the Blackborough works. He had not thought of the matter before, being only a director and the chairman, but now he had really looked into the matter and could not think how such a rotten system of charges had been brought about. He was heard to say it was up to the electricians to sell stuff at fair rates to all and that it was time they learned the effect of diversity of demand, for they had simply choked off the best part of the demand and sold below cost to the survivors.

*The highest honor man in a mathematics examination.

Preservation of Technical History Urged

Last fall, Dr. C. Matschoss, on behalf of the Verein deutscher Ingenieure, visited this country to study our technical progress in connection with his historical researches. Since his return to Germany he has addressed a communication to the American engineering societies, acknowledging his gratitude to the various societies, institutions and individuals who were of service to him in his quest for information, and urging that the societies interest themselves in collecting material for the history of American technics and industry.

After outlining his own work during the past fifteen years and the work of others in Germany and elsewhere abroad in compiling such histories as the development of the iron industry, the steam engine, the locomotive, the electrical industries, etc., Dr. Matschoss continues with the following plea for the work:

We need the history of technics in the same way as we need the history of any other human work. If we understand the great influence of all engineering work on the development of our culture and civilization—and nobody can now deny this influence, in the United States, least of all—it follows that we cannot longer neglect the history of this evolution. It is not possible to write the history of mankind and leave out the works of the engineers. Why have we no history of technics but hundreds of histories of wars, politics, etc.? The history of technics can be written only by engineers or by the help of engineers. During the last century the engineers have spent their whole time in making history and no one has had time to write it down; now it must be changed. Among the thousands and thousands of engineers we should find one or two willing to work in the historical line. And it is time to do that, for every year we lose prominent pioneers, and too often we forget with them that which they have done for mankind.

We need the history of technics also as a source of enthusiasm and idealism for our younger generation. Nothing that is great in the world is done without idealism. History shows us in the characters of the great inventors and pioneers some of the greatest idealists who have ever lived. We must tell the lives and deeds of these men to our young students. We should not forget that behind all our machines and other great engineering works made of inanimate material stand living men. The prominent engineers of today know that often it is more difficult to handle men than machines. Therefore we must learn to understand the work

of the men in the technics. Also in social interests such a history would have great value in our days.

We need the history of technics because no really educated man can be content to know a few facts only, he wishes to know the evolution. Many outside the engineering profession admire technical industry and are astonished at certain results, but they do not like technics because they know nothing of the growth of all these works, they cannot see the men who stand behind all the iron and steel. Therefore I believe that a really great history of technics is also the best means of increasing the reputation of the engineering profession.

Many ways will be possible to help forward these historical researches. It is not my task to suggest what you may do in this direction, but I trust that in America, so justly celebrated for its energy and initiative, a way will be found if there is a will.

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For Engineering Experiment Station in Ohio

The Ohio Society of Mechanical, Electrical and Steam Engineers, in view of the unquestionable advantage to the industrial and manufacturing interests of Ohio of a state engineering experiment station, recently passed the following resolutions:

Whereas, the manufacturing, industrial and engineering interests of the state of Ohio are taxed, but are not aided and fostered by any scientific and technical department of the state government, and,

Whereas, these interests form a large part of the wealth-producing and tax-paying resources of the state, and,

Whereas, many of the states of the union foster these interests by the establishment and maintenance of an engineering experiment station, therefore,

Be it resolved, by the Ohio Society of Mechanical, Electrical and Steam Engineers in session at its twenty-sixth semi-annual meeting, in the city of Akron, that this society commends the idea of the establishment of an engineering experiment station at the Ohio State University for the benefit of the people of the state engaged in the manufacturing, mineral, transportation and other industries of the state; and that we call upon the General Assembly to approve and support all reasonable efforts looking to the establishment of such an engineering experiment station; and further resolved that the members of this society as citizens urge upon the General Assembly and its individual representatives and senators the great desirability of passing a bill for the creation of the station and of appropriating adequate funds for its support.

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Forty-Fifth Anniversary of the Albany Lubricating Co.

Forty-five years ago, in 1868, the firm of Adam Cook's Sons was founded in Albany, N. Y., as the Albany Lubricating Compound & Cup Co. Its founder, Adam Cook, was a man of energy and perseverance, and the company was the direct result of conditions at that time, when oil was the only practical substance used for lubrication and as suitable means for applying it were not generally understood, it was costly and inefficient. The main drawback was that a large percentage of the oil would drip away before it had accomplished its object of reducing friction. While many engineers of that time devoted their energy to the perfection of some suitable method of applying oil to bearings, Adam Cook determined to try and make some change of the lubricant itself. He spent many days in testing out innumerable combinations of mineral and animal oils and grease until he finally produced a compound which he called "Albany Lubricating Compound." The engineers soon called it "Albany Grease" and it has ever since been known by that name.

The Albany plant was outgrown in four years and in 1872 larger quarters were secured in New York City, at 231 West St., New York City. In 1881 it was found necessary to move to still larger quarters at 313 West St. Additions were continually made to meet the growing demands for its product, but it was finally decided to concentrate the different departments and plan the manufacturing system according to modern principles and to bring under one roof all of the various departments of the business. After 30 years the West St. plant was abandoned and the present modern commodious plant at 708-710 Washington St. was placed in service.

This modern fireproof plant is a monument to Adam Cook, a man who saw a need and had the courage and perseverance to work until he had the correct solution, but more important as a reminder of the man who perfected the modern lubricating compound, is the millions of grease-filled cups in the mills and factories of the land, in the mines and on the railroads and automobiles, and finally in the air.

NEW PUBLICATIONS

STEAM TURBINES, Their Theory and Construction. By H. Wilda. Translated from the German by Chas. Salter. Scott, Greenwood & Son, London, 1912. Pages, 191; 4½x7 in.; cloth; illustrated. Price, $1.25.

The literature of the steam turbine grows apace (at least as measured by the number of volumes devoted to it). There has been enough development to furnish material for many books, each individual in its matter or method, or both. Each author must think that he has new material to present, or at least new ideas as to the manner of its presentation, but if one were to sift them for one comprehensive work he would find much that had been duplicated.

The author of the book under review spares us any prefatory excuse or explanation as to how he happened to do it. He does, however, furnish a table of the nomenclature adopted throughout the book, a practice decidedly to be commended.

The book is divided into five chapters, the first of which treats of entropy, the flow of steam, critical velocity, and classification of turbines according to the arrangement of the vanes. The second chapter deals somewhat mathematically with the flow of steam through the steam turbine. Chapter III takes up the details of steam turbines, Chapter IV their general arrangement, Chapter V condensers, and the utilization of exhaust steam, and Chapter VI, practical applications of steam turbines

ENGINEER'S HANDBOOK ON PATENTS. By William Macomber, Lecturer on the Law of Patents in the Cornell University College of Law. Published by Little, Brown & Co., Boston. Flexible leather; 270 pages 4½x7 in.; 6 illustrations; table of contents and index to sections. Price $2.50.

This is a handbook addressed to engineers, inventors and manufacturers, not so much as a treatise on the patent law as a convenient handbook of special value to those interested in deciding the essentials of a good patent and property rights in patents. The work attempts to cover the subject by giving practical answers to the questions of what a patent is, what invention consists of, what is patentable and what constitutes novelty; and to shed light upon the problems of obtaining a good patent, of knowing what a patent covers, patent litigation and property rights in patents.

Legal phraseology and terminology have been omitted as far as practicable and the style of the author is calculated to be both clear and acceptable to the average layman. A study of the book should enable the user to avoid lines of thought which have resulted in failures on the part of other inventors and enable him to secure the full benefits of a successful invention, the explanation of what is patentable being clearly detailed.

The problem of property rights is clearly set forth in the closing chapter, and, as information to inventors and engineers, is well treated by the author, who truthfully says that " it is a subject which concerns the engineer vitally and perhaps more frequently than any other in the patent law."

Statements in the text are fortified by quotations from the court decisions referred to and good indexing and paragraphing render the subjects of the work readily accessible.

FOUNDATIONS AND MACHINERY FIXING. By Francis H. Davis. D. Van Nostrand & Co., New York. Semiflexible cloth; 146 pages; illustrated; 4x6½ in. Price, $1.

This little work is at least unique in affording a separate treatise on a subject which has been strangely neglected as a specialty in engineering publications. Except for occasional articles in the technical press, there is practically no literature on the subject. The importance of good foundations for engines and machinery is so widely recognized that the author's efforts should not fail of appreciation.

The work will be found especially valuable to that class of practical engineers who are only occasionally required to decide upon provisions for engine foundations, and although the author's treatment of his subject may be disappointing to those who may seek entertainment in theoretical speculations, the latter class of readers may be awakened to prac-

tical methods of meeting problems that baffle mathematical computation.

The subject is divided into ten chapters and is treated under the heads of functions of foundations, nature of soils, designs of foundations for engines, turbines and electric generators, the materials and construction of foundations, holding-down bolts, causes and effects of vibration and methods of insulation.

The reader is furnished with 15 tables of data pertaining to the bearing power of soils, the proportions and weights of foundations successfully employed for various types of steam, gas and oil engines, dimensions of anchor bolts and plates, etc.

Although written by an English author, the work is remarkably free from terms and expressions unfamiliar to American engineers, is written in a clear style, printed in a readable type, and is well illustrated.

THE THEORY AND PRACTICE OF MECHANICS. By S. E. Slocum, professor of applied mathematics, University of Cincinnati. Published by Henry Holt & Co., New York, 1913. Pages, 442; illustrated; cloth. Price, $3.

This is intended primarily as a textbook for students in engineering colleges, and is well suited for that purpose, the theoretical principles developed being applied to the solution of practical engineering problems. It should also prove useful as a reference book for designers.

The text is divided into eight parts: Tables of mathematical and physical constants, kinematics, fundamental dynamical principles, statics, friction and lubrication, kinetics of particles, kinetics of rigid bodies and dynamics of rotation.

Under the subject of kinematics the difference between linear and angular velocity should have been pointed out, and this illustrated by application to problems on kinematic linkages. Also the kinematics of rolling cylinders and cones, parallel and straight line motions should have been included under the applications of kinematics.

Work, power, fundamental and derived units are clearly defined, and the treatment of simple structures is illustrated by well selected practical problems.

The subject of friction is especially well treated. Tables are given for the average values of the coefficient of friction for various substances and for mechanical efficiencies of various machine elements and prime movers. The average efficiencies given for electric motors and generators appears somewhat high. The part dealing with roller and ball bearings is very complete and practical, being illustrated by calculations of the size of bearings for passenger cars in railway service. Under friction transmission, paper and iron friction wheels are described and compared with toothed iron gearing.

The chapter dealing with the kinetics of particles and of rigid bodies are extremely theoretical, and most of this could well be omitted in a course on mechanics given to engineering students. The sections dealing with moment of inertia should have included solutions for sections usually used in engineering practice, as a book on practical mechanics should form the basis for the study of the strength of materials

Derivations of governor and flywheel formulas are given in a very simple and practical manner, The calculation of the weight of a flywheel for an engine capacity in kilowatts is inaccurate, the author converting indicated horsepower into kilowatts by multiplying by 0.746 without taking into consideration mechanical and electrical losses. The dynamics of the engine is well presented and the balancing of reciprocating stationary engines and locomotives is given in detail. Gyroscopic action is also considered at some length, and its application is illustrated by the monorail car and automobile torpedo.

In general, the whole subject matter of the book, and especially the part dealing with dynamics of rotation, is presented clearer and in a more definite and practical manner than in most books on the subject.

DESIGN AND CONSTRUCTION OF STEAM TURBINES. By Harold Medway Martin, Longmans, Green & Co., New York, 272 pages; 7½x10¾ in.; illustrated; cloth. Price $8.50.

This work is founded largely upon and consists in large part of articles by the same author which have had the advantage of previous presentation and discussion in the columns of "Engineering." The first chapter defines concisely the impulse and reaction types by comparison with Pelton and Girard waterwheels. Chapter II deals with guide blades and nozzles and Chapter III with their efficiency. This takes care of the steam element not only of the stator, but of the rotor of the reaction machine, leaving only impulse buckets to be treated, and these form the subject of Chapter IV. The correction for departure from adiabatic expansion, i.e., the effect of nozzle and blade efficiency upon volume and velocity is then considered.

For the purpose of reducing actual performances to other conditions of pressure, superheat and vacuum correction curves are used. Superheat corrections are practically identical for every type of turbine and over a wide range are nearly independent of the initial pressure. The gain by superheating is much greater than that theoretically due from the increased temperature of the steam. The gain from increased vacuum, however, is almost invariably less than that theoretically due, and the vacuum correction is not the same for all turbines. The effect of increased initial pressure can be reduced to a vacuum correction. If the initial and final pressures are increased in the same ratio, the efficiency of the turbine is almost unaltered. Seven correction diagrams for superheat, pressure and vacuum are given.

The succeeding nine chapters are devoted to turbine design, and Chapter XVI to the thermodynamic principles involved. The author then takes up such details as balancing; dummy and gland packings; high-speed bearings; the strength of rotary disks; and geared turbines; and steps a little aside from his subject to devote a chapter to the condenser. Illustrated descriptions are then given of the De Laval turbine; velocity compounded turbines; the compound impulse turbine; the Parsons type; disk and drum turbines, and the Ljungström turbine. This portion is accompanied by a large number of plates illustrating the various turbines in detail, and it is evidently these folding plates which have necessitated the rather more than ordinary size of volume. The book is excellently printed in large type and contains much of interest, even to the man who cannot follow the mathematical treatment of the more involved chapters.

OBITUARY

Frank P. Ohmer, chief engineer for the Indiana & Michigan Electric Co., South Bend, Ind., dropped dead from heart failure, Mar. 3, while inspecting some of the machinery at the power plant. Mr. Ohmer was well known throughout the Middle West.

PERSONALS

Sanford H. Smith, formerly of the Allied Publishing Co., has become associated with the "Isolated Plant Magazine," New York City, in the capacity of managing editor.

A. M. Houser, mechanical expert for the Crane Co., Chicago, Ill., although busy in connection with the Bridgeport, Conn., branch of the company, found time to visit this office a few days ago.

G. G. Ross, of Ross & Taggert, consulting engineers, 114 Liberty St., New York City, has returned to the home office after a year's absence. Mr. Ross has been reorganizing a large manufacturing plant.

W. A. Ostrom, chief electrician in the Saskatoon (Sask.) power house for the past year, has resigned to enter again the employ of the Canadian General Electric Co. as superintendent of construction in the West.

C. R. Smith, designer of stations for the General Electric Co., who has been in charge of the reconstruction work of the Newport News & Old Point Railway & Electric Co., at Hampton, Va., has returned, having completed the rebuilding of the station and ice-making plant.

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An amusing instance of undue prejudice in regard to the goods of different manufacturers was recently quoted in connection with the working of the recent act for the suppression of illicit commissions. A certain engineer-in-charge praised some lubricating oil supplied by A and condemned the oil supplied by a competing firm B. The plant owner, suspecting the existence of some outside influence, secretly changed over the contents of the oildrums submitted by the different firms and asked the engineer to give both oils a further trial. In ignorance of the substitution, the engineer continued to give unqualified approval to the contents of A's drum, and similar condemnation to the oil in B's drum. We do not know what happened to the engineer, but he no doubt learned a very useful lesson. There is, of course, a moral for the employer, and that is, the more poorly paid a servant is, the more susceptible he is likely to be to financial inducements to betray his trust. In theory this argument should not apply, but human nature is a very difficult thing to bring closely into line with theory.—"The Power User."

Bituminous Coal Sizes

Relative to bituminous coal sizes the Bureau of Mines informs us as follows:

Bituminous coals are not prepared by systematic sizes. In all of the bituminous coal fields a great deal of coal is shipped as "run-of-mine." In the coal fields of Pennsylvania, Ohio and West Virginia, a considerable part of the coal is merely screened through bars about 1½ in. apart. In the Middle West the coal is usually screened over shaking screens with round holes. The coal passing through a 6- or 8-in. diameter hole and over a 3- or 4-in. hole is called small or railroad lump; the coal through a screen with 3-in. diameter holes and over 1½-in. or 2-in. holes may be called "egg" coal; between 2½ and 1-in. may be termed "nut" coal; between 1- and ½-in. may be termed "pea" coal and below that, "slack or duff." There are no precise grades and there are all kinds of mixtures, so that in buying it is always customary for the buyer to specify that the coal shall be passed over and through certain size screens; bar screens or shaking screens as the case may be.

There is a little better uniformity in washed coal of the West, but even these vary from the so called standard sizes which range from "nut" to "duff" but are always specified as washed coal.

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Washington's Great Undeveloped Water Power

The Cascade Range of Washington and Oregon has an inexhaustible store of running water. In the opinion of the hydrographic engineers of the United States Geological Survey few areas in the United States present more favorable opportunities for the development of water power.

The crest of the range is 6000 to 8000 ft. above sea level, and the higher peaks extend into the region of everlasting snow. From Mount Rainier in Washington to Mount Shasta just south of the Oregon-California line, both more than 14,000 ft. in elevation, many snow-covered peaks are the perpetual fountains that supply the flow of strong rivers running westward to the coast. The precipitation of the area is heavy, the Cascade Range receiving about 100 in. of rainfall a year.

The streams of the Cascades possess in high degree the requisite characteristics for the successful development of water power, namely, steep gradient, abundant water, and comparatively uniform flow. Add to these the enormous resources of timber, metal and other minerals, and lands as yet developed only in small part but promising an extensive future market for water power, and it becomes apparent that the large areas of country surrounding the Cascade Plateau have in the undeveloped power of their streams very great potential wealth.

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Electrification of London's Suburban Railways

The London, Brighton & South Coast Ry., which since 1909 has electrified some 70 miles of its suburban lines, recently decided upon a further electrification, to include, when completed, at least 170 miles of additional trackage, says a consular report. The work will extend over four years. The lines to be transformed will be divided into sections.

The whole of the company's suburban system, in the area extending from London to Croydon, Purley and Coulsdon, and from London to Sutton and Cheam, is to be electrified. This action has been decided upon in view of the satisfactory results of the company's electric service now in operation. This road alone of the railways of London which have adopted electrification is committed to the single-phase, overhead, alternating-current system, and will employ it in its extension. Two of the principal reasons for continuing the system are that it is not desirable to have a third rail and that it will be necessary, in the near future, to extend the electrification to Brighton and Eastbourne.

The company purchases its current from the London Electric Supply Corporation. In consequence of the increased demand, the latter company will be compelled to add materially to its present plant, thus making it one of the largest in England.

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Glibly as everyone now talks of horsepower there are few things concerning which the average man knows so very little as he does of what strength the human or the animal is capable of exerting.

Some idea of this has come to light through a series of interesting tests recently made to determine the respective pulling power of horses, men and elephants. Two horses weighing 1600 lb. each, together pulled 3750 lb. or 550 lb. more than their combined weight. One elephant weighing 12,000 lb. pulled 8750 lb. or 3250 lb. less than its weight. Fifty men, aggregating 7500 lb. in weight, pulled 8750 lb., or just as much as the single elephant; but, like the horses, they pulled more than their own weight. One hundred men pulled 12,000 pounds.

NEW EQUIPMENT

ATLANTIC COAST STATES

The installation of a water-works system at East Long Meadow, Mass., is under consideration. F. C. Burton is chairman of the board of selectmen.

Plans are being prepared for the construction of a power plant at Clayville, N. Y., to be erected for H. D. Wood, of Phoenix, N. Y.

The installation of new equipment in the municipal electric-light plant at Jamestown, N. Y., has been recommended by C. O. Johnson, superintendent of the plant.

The power plant of the Jamesburg Light & Power Co., Jamesburg, N. J., was destroyed by fire recently. The plant will be rebuilt at once. Estimated loss, $10,000. Frederick Greenslade is superintendent.

SOUTHERN STATES

Stanley Scott is interested in the project to install an electric-light plant at Eastville, Va.

An election has been called for Apr. 15 at which the citizens of Wilson, N. C., will vote on the proposition to issue $80,000 in bonds for the improvement of the municipal electric-light plant.

Plans are being prepared by J. Newton Johnson, Florence, S. C., for the construction of a water-works system at Manning, S. C.

The citizens of Abbeyville, Ga., will hold an election on Apr. 8 to vote on the proposition to issue bonds for $8000, for the construction of a municipal electric-light plant.

Plans are being prepared by the J. B. McCrary Co., Atlanta, Ga., for the installation of an electric-light plant and water-works system at Marshallville, Ga. Estimated cost, $20,000. J. O. Boonton is mayor.

An election will probably soon be called to vote on a bond issue for the installation of an electric-light plant and water-works system at Lake Butler, Fla.

Bids are being received by the authorities of Micanopy, Fla., for equipment for the electric-light plant, including a 75-kw., 2300-volt, 60-cycle, three-phase generator, with switchboard, instruments, exciter, noncondensing, four-valve steam engine, 100-hp. boiler with piping, feed-water pump; engine to be directly connected to generator.

CENTRAL STATES

An expenditure of $16,000 is planned by the Thornapple Gas & Electric Co., for the improvements of its power plant at Hastings, Mich. Lewis W. Heath, Hastings, is manager.